POWER

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His Highness

By Berton Braley

Down below the sidewalk level, in a spot remote, obscure,
Where the light is artificial and the ventilation's poor,
Where the air is moist and heavy and the pistons hiss and throb
With the labor they are doing, you will find him on the job.
He's the power dignitary,
He's the potentate down here,
He's the royal stationary
Engineer.
Give a cheer
For the very wise and wary
Engineer!

In his overalls and jumper with a waste ball in his hand,
He stands among his engines with an air of calm command,
And now and then he polishes and rubs 'em briskly, too,
To keep 'em bright and shining for the work they have to do.
You can bet that he is jerry
To each working part and gear,
Is the canny stationary
Engineer.
Give a cheer
For the very wise and wary
Engineer!

If he should shirk his duties, it would disarrange the scheme,
He's the man behind the lighting and the man behind the steam,
He's the man behind the engine in a kingdom of his own,
He's the minister, the monarch and the power behind the throne.
If he ever grew con-*trary*,
He could tie us up, that's clear,
Could this potent stationary
Engineer.
Give a cheer
Just a cheer to keep him merry
In the work that he must carry.
Here's to Mr. Stationary
Engineer!

New Cincinnati Hospital Plant

By Thomas Wilson

SYNOPSIS—A modern direct-current plant recently completed to supply a hospital with heat, light, power, steam and refrigeration. The boilers are equipped to burn natural gas or coal. Live and exhaust steam heaters are used in connection with a forced-circulation hot-water heating system. Unusual provisions are made in steam and water piping to insure continuity of service.

The new Cincinnati General Hospital, which is being built at a cost already approximating $4,000,000 to take care of the city's sick poor, will, when the present plans are fulfilled, be the largest and best-equipped of its kind in the world. The site covers 65 acres, and the 24 completed buildings, ranging in height from one to five stories, occupy the southeast 27 acres of this. Eleven wards are now in use, and eight more will be built, besides a large medical college for which the foundation has been started recently. Last year about 8,000 patients were admitted, and at present the average population per day runs close to 485. The annual maintenance cost is $362,000. The cost per patient per day averages $2.

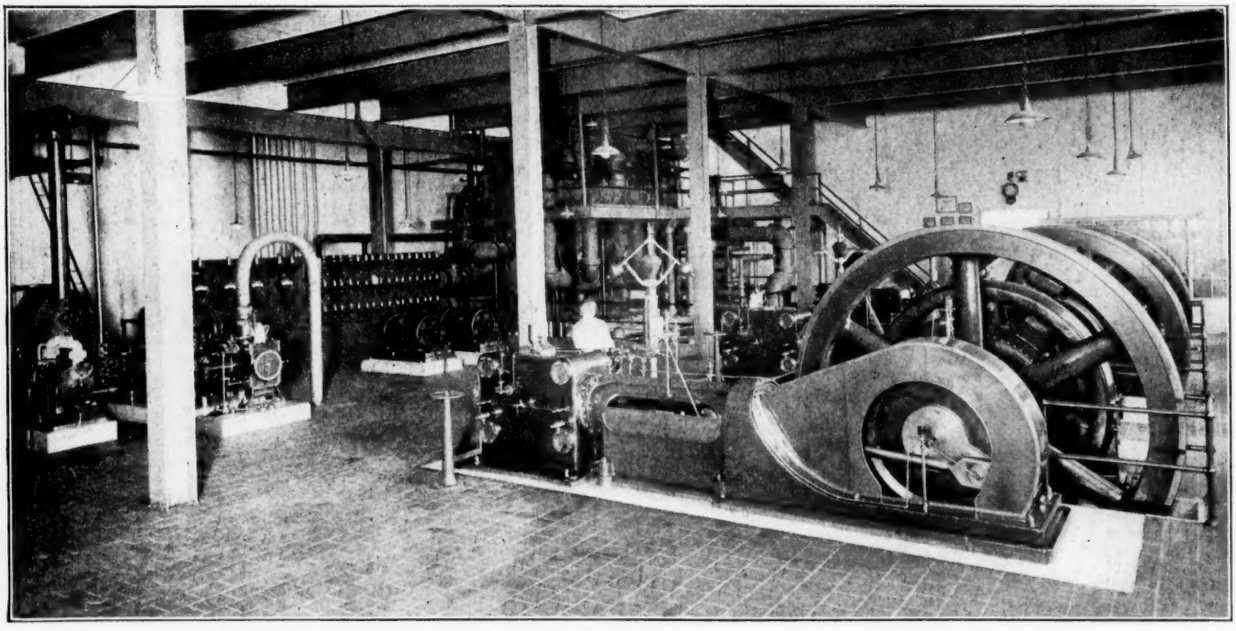


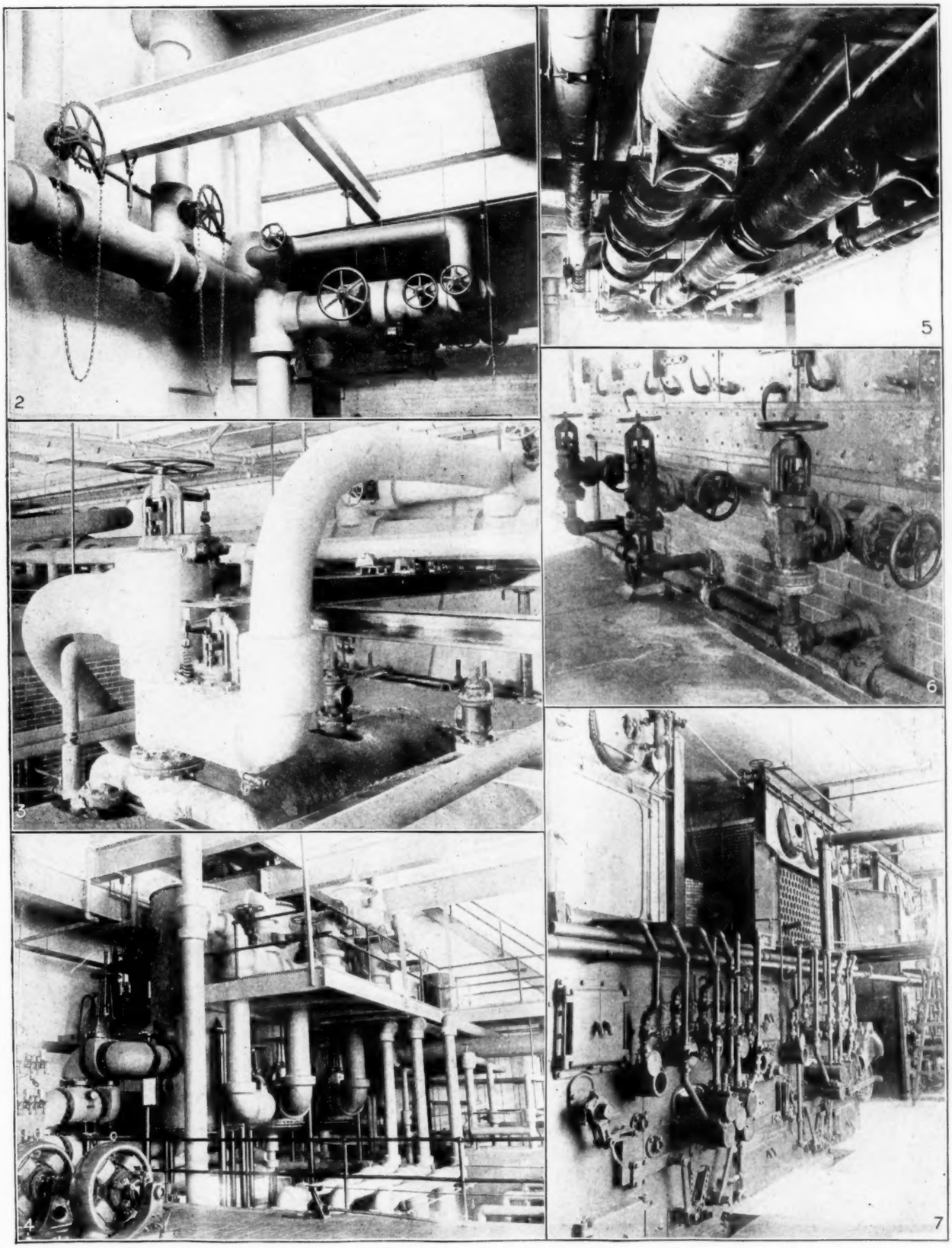
FIG. 1. A GENERAL VIEW OF THE ENGINE ROOM OF CINCINNATI HOSPITAL PLANT

All these buildings are heated by a hot-water system, selected principally because of the large area covered and the desire for uniform heat. Forced circulation is employed, and heaters for exhaust and live steam are located in the engine room. The exhaust back pressure does not exceed one pound. Should the supply be inadequate and it is not desired to use the live-steam heaters, provision has been made to supplement the exhaust with live steam at reduced pressure in the exhaust-steam heaters. Steam at 40 lb. pressure is supplied to the kitchen and for sterilizing purposes in the wards, the reduction from boiler pressure being made at the point of using. There are also connections for snow melting on the ward roof-gardens, and live steam at boiler pressure is required in the laundry.

Current is generated at 220 volts and distributed on the three-wire plan by the use of two balancer sets having a capacity equal to 20 per cent. of one generating unit. The load runs close to 400 amp. and is practically constant, as the motor load during the day offsets the larger lighting load at night. With the completion of the new medical college and the addition to the institution planned, the electrical load will be greater and more in conformity with the capacity of the generating units installed. There are three 250-kw. units, Fig. 1, and one of them running at 40 per cent. of rating is sufficient to carry the load. On account of this light load it was desirable to drop the steam pressure from 125 lb. to 100.

More or less refrigeration is required, and the hospital uses several tons of ice per day. All of the buildings are mechanically ventilated, and compressed air is supplied throughout the institution for various purposes. The services are all maintained by a central plant of modern design planned in particular to guard against shutdowns. The boilers are equipped to burn natural gas or coal as fuel, and in the steam and feed-water piping unusual provisions have been made to keep the plant running in case of accident to any particular line.

The boiler room contains three water-tube boilers each with 5,200 sq.ft. of heating surface and 84 sq.ft. of projected grate area in twin furnaces of the side-feed type. There are also two 260-hp. boilers that were transferred from the older hospital. Each of these has 2,600 sq.ft. of heating surface and a single side-feed furnace having a projected grate area of 42 sq.ft. The actual grate area for the larger boilers is 104 sq.ft. and for the smaller boilers 52 sq.ft., giving a ratio of heating surface to grate area in either case of 50 to 1. The stokers are installed in full extension dutch-oven settings, and to conform



FIGS. 2 TO 7. VIEWS OF VARIOUS FEATURES OF INTEREST IN THE CINCINNATI HOSPITAL PLANT

Fig. 2—Chain-controlled valves to high-pressure heaters and reducing valves to exhaust-steam heaters. Fig. 3—Nonreturn stop valves on 260-hp. boilers. Fig. 4—Exhaust-steam heaters in engine room. Fig. 5—Heating and service supply pipes in main tunnel. Fig. 6—Boiler blowoff valves. Fig. 7—Boiler fronts equipped with gas burners

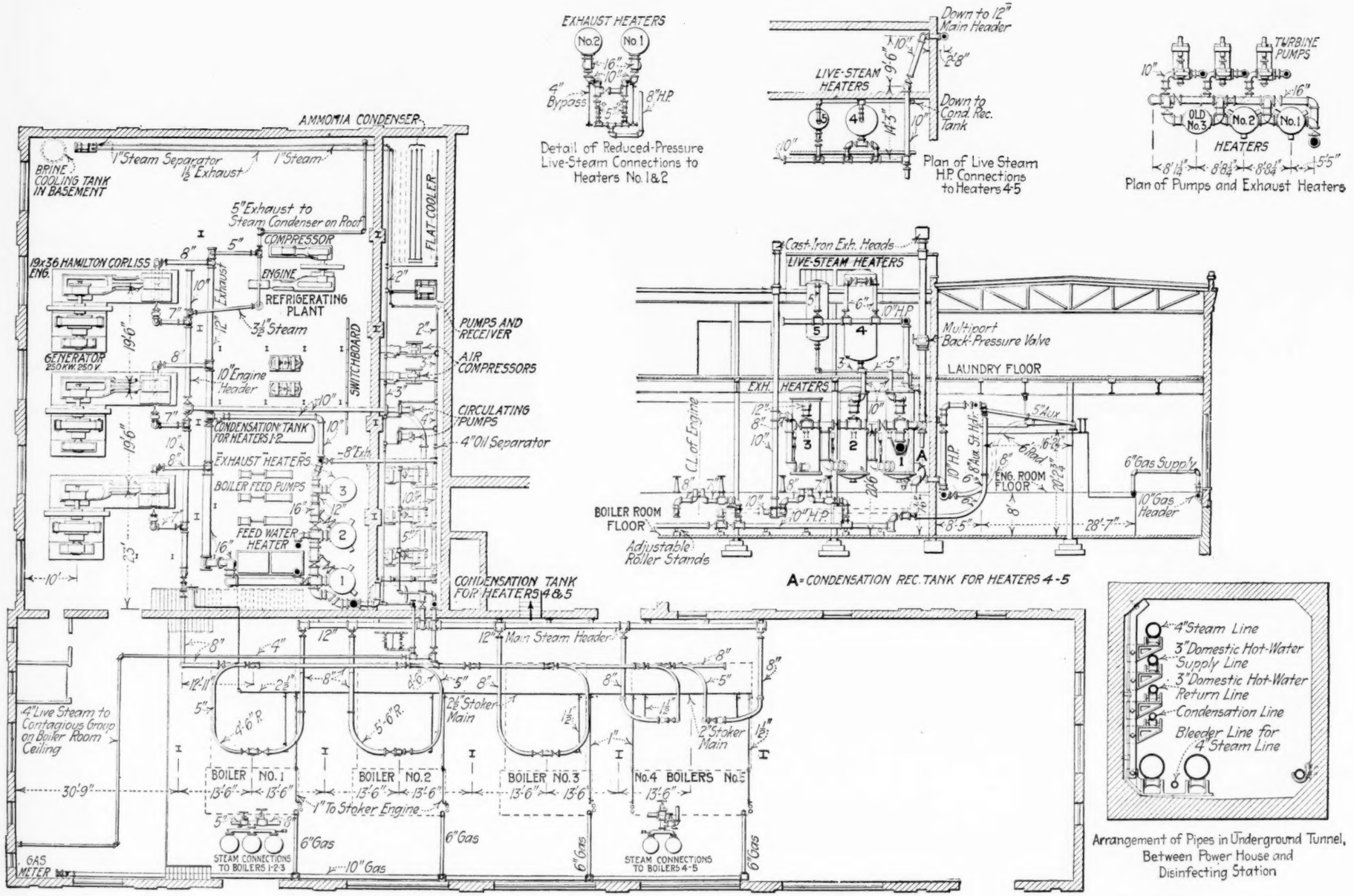


FIG. 8. PLAN OF STEAM PIPING IN ENGINE AND BOILER ROOMS, PIPING TO HEATERS AND DETAILS OF PIPE SUPPORTS IN TUNNEL

with the smoke ordinance of Cincinnati, each furnace arch extends 2 ft. beyond the rear line of the grate bars. This additional length of arch is provided to heat the volatile gases from the coal at the rear of the furnace to the point of ignition, so that more efficient combustion and smokeless operation are insured.

Although this provision was made for coal-burning, this fuel is used only a few weeks in the year. The plant is situated where natural gas is available at a price at least equivalent to that of coal and has the advantage of eliminating dirt and dust, which is desirable in the vicinity of a hospital. To provide for the gaseous fuel, the at different levels so that they might enter the stack one under the other, or in other words, converge into one after they had rounded the right-angle turn into the stack, Fig. 9. In this way the gases from either leg do not interfere with the others to cause a drop in draft. The elliptical opening into the stack has an area of 54 sq.ft., while the area of the stack itself at the top is 50 sq.ft. Per 1,000 sq.ft. of heating surface in the boilers there is 2.6 sq.ft. of breeching, 2.4 sq.ft. of stack area and 20 sq.ft. of connected grate area.

From the rear of the boiler setting to the boiler-room wall the distance is 8 ft. 6 in. The blowoff line, Fig. 6,

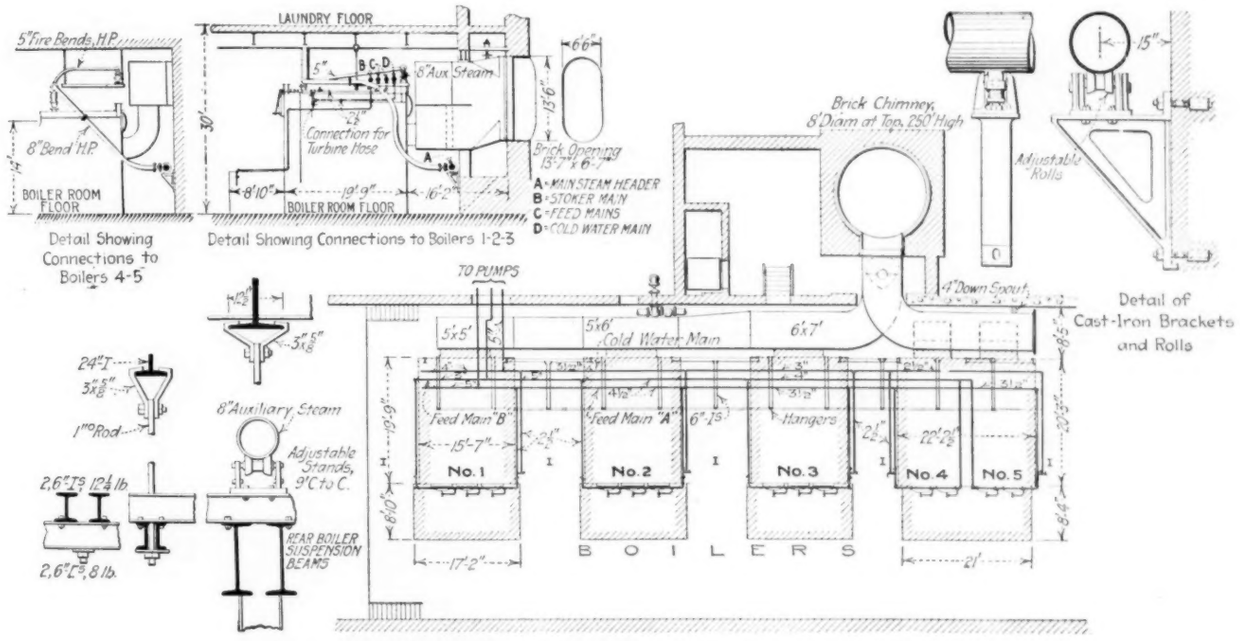


FIG. 9. FEED-WATER PIPING AND BREECHING ARRANGEMENT

larger boilers have twenty 5-in. burners and the smaller boilers ten burners of the same size. Through 1½-in. pipes these burners are supplied from a 10-in. gas main running across the boiler fronts, Fig. 7. In changing fuel it is only necessary to break the joints in the gas-supply pipes, pull out the burners and remove the brick covering from the grates. As the use of coal is limited, no special provisions were made for handling this fuel. It is wheeled up an incline and along the top of the furnaces and dumped directly into the stoker magazines. An additional force of three firemen and nine laborers to handle the coal is required. On this account principally, natural gas has been found to be the cheaper fuel and has been used in the furnaces with excellent results.

The boilers are vertically baffled and of the three-pass type. They are supported by steelwork independent of the brick walls of the setting. From a rear outlet the gases discharge into a rectangular breeching resting on steel I-beams supported by the boiler-room wall and the vertical columns in the boiler settings. Where there are no boilers, the breeching is suspended from the ceiling by rod-and-beam construction.

The radial-brick stack, which is 8 ft. in diameter at the top and 250 ft. high, is located centrally, with the three larger boilers on one side and the two smaller ones on the other. Instead of coming together in the horizontal run back of the boilers, the breechings were erected purposely discharges into a large tank which has a water-sealed connection to the sewer and a vent on top to allow the steam or vapor to escape.

The engine room, which is in one leg of the L-shaped building containing the power plant, is on a level 8 ft. above the boiler-room floor. Consequently all of the piping is carried underneath the floor, and the units, except the ammonia-compressor engine, all have bottom connections. As the total maximum load was estimated at about 500 kw., three Corliss engines, 19x36 in. and 120 r.p.m., with double-ported valves and double eccentrics, directly connected to 250-kw. 220-volt two-wire direct-current generators, are installed. Although the steam rate is up to the standard for units of this type, there was no occasion to choose engines from the standpoint of low steam consumption, for with a large heating load and a continuous demand for large quantities of hot water, all the exhaust steam is utilized the year around.

Two balancer sets, rated at 25 kw. each, permit of three-wire distribution, the motors being served at 220 volts and the lighting at 110 volts. The nine-panel switchboard is of standard construction and has the usual complement of static instruments.

To supply the hospital with ice and to cool 14 refrigerators for kitchen service, a 30-ton compression and brine system was installed. The high-pressure end consists of a 10x18-in. compressor having a maximum speed of 90

r.p.m. and is driven by a 12x30-in. Corliss engine of the same type used to drive the generating units and a two-pipe ammonia condenser. These are installed in the engine room. The brine and ice tank and the ice-storage room are located in the kitchen and dining hall, some 300 ft. distant from the plant. It was considered more convenient to install the ice tank near the point of consumption and to convey the ammonia and distilled water from the plant rather than to locate all of the apparatus close together.

The ice tank contains ninety 300-lb. cans, and the brine for the refrigerators is circulated from this tank by motor-driven pumps. The agitator in the brine tank is belted to a motor, and to guard against a shutdown without the knowledge of the power-plant operators, a battery-and-bell signal system is used. A rudder is inserted in the tank just in front of the agitator, and as long as the brine is in motion it is maintained in a position parallel to the flow. If the flow should stop, a spring pulls the rudder out of position and an arm at the top closes the electric circuit, ringing a bell in the engine room and notifying the attendants. A calcium-chloride solution is used, and its temperature in the refrigerators ranges from 16 to 24 deg. Other equipment in the engine room consists of two steam-driven air compressors supplying air at 65 lb. pressure for operating laundry machinery and at reduced pressure for medical purposes and thermostatic control.

As a discontinuance of service from the power plant would be serious to the hospital, exceptional provisions were made in the piping to guard against shutdowns due to a rupture of any part of the live-steam and exhaust mains. The high-pressure piping is full-weight wrought-iron pipe with extra-heavy fittings. The flanges are screwed on, refaced and peened. In the boiler room a 12-in. main header and an 8-in. auxiliary header have been provided. Through nonreturn boiler stop valves of angle or globe type there is a connection from each boiler to either header, Fig. 3. The pipe sizes are 8- and 5-in. respectively, and their arrangement is shown in Fig. 5. The main header is supported on cast-iron brackets having adjustable roller stands. It supplies primarily the main generating units and the ammonia-compressor engine. From the center of the main a 10-in. connection leads to the engine-room main, which is carried under the floor on adjustable roller chairs set on concrete piers. From this main 7-in. connections lead to each engine, with double spring joints and a separator in each line.

The auxiliary header, which normally supplies steam to the pumps and the contagious disease group of buildings, is also connected through a fire bend to the end of the engine header. It is thus possible for either boiler header to be down and still supply the prime movers.

There is also an independent 2½-in. stoker engine main connected to the steam piping on the boiler side of the stop valves. From this main a double-valved branch leads to each stoker engine, so that any one of the engines can be operated whether its respective boiler is under steam or not. The stoker main is cross-connected to the feed-pump main, and if all of the boilers are cut off from both the 12- and 8-in. headers, sufficient steam will still be supplied from the stoker header to operate at least one boiler-feed pump.

PRINCIPAL EQUIPMENT OF CINCINNATI GENERAL HOSPITAL PLANT

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
3	Boilers	Horizontal water-tube	520-hp.	Generate steam	100 lb. pres., stokers, natural draft	Babcock & Wilcox Co.
2	Boilers	Horizontal water-tube	260-hp.	Generate steam	100 lb. pres., stokers, natural draft	Tudor Boiler Co.
8	Stokers	Side-feed	42 sq.ft. projected area	Serve boilers	Double stokers under larger boilers	Murphy Iron Works
1	Stack	Radial brick	8-ft. diam., 250 ft. high	Serve boilers		Heine Chimney Co.
5	Efficiency meters			Measure drop in draft		W. A. Blouck & Co.
80	Burners	Gas	5-in.	Serve boiler furnaces	Burn natural gas	Gwynn Gas Burner Co.
3	Pumps	Single-cyl., double acting	12x8x16-in.	Boiler-feed water		Union Steam Pump Co.
1	Heater	Sorge-Cochrane open	2,500-hp.	Heat feed water	Exhaust steam; combination heater and softener	Harrison Safety Boiler Works
3	Engines	Hamilton Corliss	19x36-in.	Main generating units	100 lb. steam, 1 lb. b.p., 120 r.p.m.	Hooven, Owens, Rentschler Co.
3	Generators	Direct-current	250-kw.	Main generating units	220-volt two-wire	(Fort Wayne) General Elec. Co.
2	Balancer sets		25-kw.	Permit three-wire distribution		(Fort Wayne) General Elec. Co.
1	Compressor	Ammonia	30-ton, 10x18-in.	Refrigerating system	Max. speed, 90 r.p.m.	Triumph Ice Machine Co.
1	Engine	Hamilton Corliss	12x30-in.	Drive compressor	Max. speed, 90 r.p.m.	Hooven, Owens, Rentschler Co.
1	Refrigerating system	Compression, brine	30-ton	Ice-making and refrigerating		Triumph Ice Machine Co.
2	Pumps	Centrifugal		Circulate brine	Motor-driven	Triumph Ice Machine Co.
2	Pumps	Automatic	5x3x6-in.	Distilled water to ice tank		Union Steam Pump Co.
2	Air compressors	Steam-driven	8x10x8-in.	Compressed air for general use	75 lb. air, 116 cu.ft. free air, 160 r.p.m.	Union Steam Pump Co.
3	Heaters	Exhaust-steam	Care for 300,000 sq.ft. direct radiation	Raise temperature of water in heating system		Almirall & Co.
2	Heaters	Live-steam				Almirall & Co.
3	Pumps		10-in., 3,000 gal. per min.	Circulate water in heating system	Driven by 24-in. turbine	Alberger Pump & Condenser Co.
2	Heaters	Turbo-volute, domestic	5,000 gal. per hr.	Heat water for domestic use		Alberger Pump & Condenser Co.
1	Heater	Domestic	4,000 gal. per hr.	Heat water for domestic use		Alberger Pump & Condenser Co.
2	Pumps	Turbo-volute	3-in., 250 gal. per min.	Circulate domestic hot water	Steam-turbine driven	Alberger Pump & Condenser Co.

Fort Wayne switchboard; General Electric static instruments; Cochrane steam and oil separators; Sorge cast-iron exhaust heads; Lunkenheimer stop valves, blow-off valves and all steam and water valves; Lunkenheimer "Marvel" mechanical oil pumps; Anderson steam traps; Hughson pressure-reducing valves and pump regulators; Kitts water-level regulators; Cochrane Multiported back-pressure valves; Thomas & Smith "Acme" air washers; Sturtevant fans; American Radiator Co.'s cast-iron vento coils.

The engine exhaust main, which is 12 and 15 in. diameter, first leads to a combination feed-water heater and water softener which has a capacity of 60,000 lb. of water per hour. The heater may be bypassed, leaving the oil separator in service. The exhaust piping passes on and is connected to each of the three exhaust-steam heaters, so that the steam may pass through them in series or in parallel, or be shut off from any one of the three. The water connections to the heater are arranged in the same way, and the flow is counter-current. From one end of the heater main the exhaust piping leads to a 14-in. outboard exhaust having a multiported back-pressure valve and topped by a cast-iron exhaust head. The other end of the main leads to a 10-in. free exhaust and also to the center of the engine exhaust main. It is thus possible for the steam to come either way to the heater. An 8-in. pipe receives the exhaust steam from the two distilled-water pumps belonging to the refrigerating system, the two air compressors, the turbines driving the domestic hot-water circulating pumps and the three large circulating pumps on the heating system. This exhaust main

connects with the domestic heaters and passes on to the main outboard exhaust. Temperature-regulating valves control the supply of steam to each domestic water heater. The exhaust pipe from the ammonia-compressor engine is connected to the steam condenser of the distilled-water system and also has a crossover to the main-engine exhaust line. The stoker engines exhaust into the ashpit.

The three exhaust heaters, Fig. 4, and the two live-steam heaters have been designed to care for the equivalent of 300,000 sq.ft. of direct radiation. At present they serve 90,000 sq.ft. of direct surface, 15,000 sq.ft. of pipe coils and 12,600 sq.ft. of indirect vento fan coils, all of which is equivalent to 185,000 sq.ft. of direct cast-iron radiation. All of the buildings are ventilated, the vento-coils being served by hot water. Air washers are provided, and the fans are all motor-driven.

With insufficient exhaust steam to keep the water of the heating system at the proper temperature, the live-steam heaters may be called into service. A 10-in. connection from the main boiler headers is carried across the front of the two live-steam heaters. From this line two 6-in. branches with globe valves having sprocket and chain control, Fig. 2, lead up through a tee to the largest of the two heaters. A 5-in. branch, also having a chain-operated globe valve, leads to the smaller heater.

From a tee on the end of this 10-in. main a live-steam connection leads to two special reducing valves and to the bypass connections around them. The piping for these reducing valves is cross-connected so that either or both may be used, and the reduced-pressure steam is conducted to two of the exhaust heaters on the floor below through 10-in. pipes. It is thus possible to cut out the live-steam heaters entirely and supplement the exhaust with sufficient live steam to meet the demands.

There are three separate boiler-feed lines carried over the tops of the boilers on adjustable roller chairs resting on steel supporting beams, hangers, etc., Fig. 9. One line is for the water from the feed-water heaters, one for live-steam condensation from the live-steam heaters or from the exhaust heaters when they are being operated with reduced-pressure steam, and one for cold water for operating the turbine-flue cleaners or for emergency when one of the other lines might be out of service. These three lines are carried back to the water end of the feed-water pumps, and each discharge pipe at the pump has a connection to each pump governor, so that the pumps may be controlled from the boiler room. The condensate from the exhaust-steam heaters is carried to the feed-water heaters or bypassed to the sewer. There are three boiler-feed pumps of the single-cylinder double-acting outside center-packed plunger type. The steam supply is taken from the pump main, which is continued past the pumps and connects to the float control valve on each of the two condensation tanks, one for the live-steam heaters and the other for the exhaust-steam heaters. From each of these control valves a branch joins a loop header leading back to each steam supply to the pumps. This header is valved so that any of the pumps may be controlled from the discharge line or automatically started by a change of levels in the condensation tanks. Fig. 8 is a plan of the steam piping in the engine and boiler rooms.

It was planned to feed the returns from the high-pressure heaters continuously into the boilers, the balance coming from the feed-water heater under the control of the pump governors. At present the live-steam heaters are not in operation, so that all condensation is brought back to the feed-water heater and returned to the boilers. If for any reason the stop valve at the boiler on the line from the condensation tank serving the high-pressure heater should be closed, the pump governor would shut down the pump. It would then be possible to use the bypass around the governor and control the feed by the regulator on the tank. If the stop valve on the other hot-water line should be closed, a relief valve allows the water to discharge into the blowoff system. These are precautions designed to prevent rupture of the piping and interference with continuous operation of the power plant.

A. G. Hall is mechanical director of the hospital and is in charge of an operating force consisting of a chief engineer, three operating engineers and three oilers—one on each 8-hr. shift—and one boiler cleaner. As previously stated, when coal is burned, the force is increased by three firemen and nine laborers. Dr. A. C. Bachmeyer is superintendent and medical director and Dr. Walter E. List, assistant superintendent. George F. Wright, mechanical engineer for Samuel Hannaford & Sons, architects for the entire institution, was responsible for the design and complete layout of the power plant.

Power Situation in Germany and Austria During the War

Special Correspondence

SYNOPSIS—Increased demand for power in munitions manufacture has been met by curtailing demands in other fields. Power supplies have been mobilized, and in many instances the copper in feeders and machines has been utilized for war purposes.

The outbreak of the war and the disorganization of industrial enterprise following this event had a decided influence on power production and consumption all over Germany and Austria-Hungary. When war was declared and the government called in the military reserves, a great number of industries were suddenly deprived of their manual help and consequently were compelled to reduce production. Also the power industries, especially the central stations, were suffering from shortage of labor, and the immediate result was a general curtailment of service. This reduction in industrial activity all over the country, however, made the effects of power supply less felt, and the fact that just then the electric-light supply had to meet only the summer demand helped the central electrical supply stations to tide over their initial difficulties.

More difficult to deal with was the disorganization in the demand for power. While in normal times this demand is spread fairly evenly over the whole territory and the larger overland stations consequently can easily arrange their output, the demand ceased in certain districts with the outbreak of the war, while it became unexpectedly large in others. In sections where small industries prevailed there was little demand for power, while everywhere in the iron and steel and the machine-building industries the demand rose to such an extent that the existing stations were taxed to full capacity. New industries springing up as a result of the war, especially such products as had formerly been imported, soon brought new problems, which had to be solved at short notice. Later, the problem of raw materials and fuel added to the many difficulties under which power production had to be carried on.

All these problems were met as they arose, either by reorganization or by complete changes, and at the end of 1915 conditions had accommodated themselves to the war so that a fairly normal, even if not entirely satisfactory, operating system had been created.

Adjustment of Tramway Load

The first months of the mobilization of the army, while taking away manual help from the stations, found added work for all those power stations that had to supply railroads and tramways, especially in those districts which were mostly affected by the mobilization and the carrying of troops. For instance, in the Rhineland, which is covered with a network of electric railways, a great demand for tramway facilities arose, making necessary the employment of all available rolling stock and also power. Not only were the tramways used largely for the transportation of troops, but they served also as feeders for the ordinary railways connecting the armament plants with the western front. As the system of power stations working in that neighborhood is closely interrelated, it was easy to organize the power supply in such a way that no difficulties arose. In all those districts the power supply to railways increased, but it became necessary to cut down tramway facilities in other directions, and nearly all the street-car systems in the larger cities began to curtail their service, thus making possible the employment of the stock in other places and lightening as well the work of the generating stations. Owing to the lack of men for running the cars as well as travelers to use them, this reduction could be carried out without undue hindrance to the service, which has been everywhere sufficient for the decreasing demand.

In some cases where there was competition between existing stations and overlapping, this competition ceased to exist. In Vienna, for instance, a considerable saving was gained by simply closing down the plant of the Allgemeine Oestereichische Elektrizitätsgesellschaft and coupling the whole service onto the Wiener Städtische Elektrizitätswerke, which not only brought about cheaper operation of the service, but made the supply more effective.

With the progress of the war the shortage of labor in the power stations was somewhat relieved by measures taken by the government. Many industrial plants were closing down, and labor was set free which could be used somewhere else. Further, female labor, which formerly had been employed only in certain industries, began to spread, and we find machine shops, ammunition plants and other industrial enterprises employing a large number of women. In the central stations the shortage of skilled labor was not so easily remedied. By coupling together existing plants as previously mentioned, a saving was gained, but this was not sufficient, especially as power generation very soon became an important factor in industrial preparedness. Therefore power plants began to ask exemption for their leading employees, which was granted where necessary. The main difficulty was that the government itself became a large employer of skilled electrical labor, having many uses for such workmen in the field. The telephone plays an important part in military operations, trenches are lighted electrically, and electricity is used to charge the wire entanglements before the trenches. Nevertheless, the generating stations were able to secure sufficient assistance to keep their plants going, even if curtailments of power and light service became necessary.

Electrical Plants Closed Down

In the middle of the year 1915, when it became evident that Germany and Austria would have to take steps to relieve the shortage of raw materials such as copper and nickel, which had formerly been imported, the closing down of electrical plants became more general. Where possible, concentration of power generation was resorted to, so as to set free the copper used in machinery and cables. In this way both countries were able to gain considerable quantities of copper. In Vienna, for instance, 480 tons of copper is said to have been recovered by using the cables of one of the closed plants. Moreover, in building new machines an effort has been made wherever possible to replace copper by iron or other metals.

However, it seems that the ordinary means of obtaining the desired metals were not sufficient, and in the autumn a new order appeared, making compulsory the registration of all kinds of unused electrical machinery. The committee that had this work in hand had authority either to destroy the machines so as to gain the copper and other desired metals or to order their rebuilding after plans that would recover as much of those metals as possible.

Handling Rebuilt Machines

The rebuilt machines were taken over by the distribution office for electrical machinery of the War Ministry, which acted as a kind of exchange, facilitating the distribution of all kinds of electrical machines to consumers who could make use of them. This system of machinery distribution has been applied to other machines, and it was possible to relieve in this way any arising shortage on one side without necessitating the building of new machines which, under the existing circumstances, would have been costly and have reduced still more the existing stocks of raw materials. The rebuilding of old machines and the construction of new ones without the use of copper was not always easy, and great difficulties were met in many instances. It is likely, however, that the knowledge thus gained will have some influence on the construction of power machinery in the future.

Another effect of the war on the electrical-power industry both in Germany and Austria has been curtailment

of building activity, shortage of money for private and public enterprises having made impossible industrial expansion.

Coal was obtainable in sufficient quantity and for reasonable prices, but there existed some difficulty in shipping it to the spots where there was demand. There was also a shortage of oil for the Diesel motors, which was aggravated for some time, especially in Hungary, through the occupation of Galicia by the Russians. When the latter were driven out of the province, they destroyed much of the industrial property and set fire to several of the oil wells; but it seems that the electric-power stations in this district have fared rather well, and little of their property was destroyed.

Although lack of coal was felt less in Germany than in Austria, in the former country transportation facilities were somewhat irregular and generating stations were compelled to carry larger stocks than usual so as to meet any sudden stoppage. This has reopened the question of making a better use of the existing water reserves for generating purposes. Generally it seems, however, that Germany's water reserves will be of better use to industries in the form of an inland water-transportation system than for power generation. The present project of creating a government monopoly in power generation foresees the use of the now practically valueless peat reserves.

Germany and Austria-Hungary will have to economize, and they cannot economize better than by gaining the same results by the employment of cheaper means. So it is likely that many of the stations, having reduced their working expenses to the lowest possible standard, will continue their work under the same economic conditions after the war. This will set labor free in other directions.

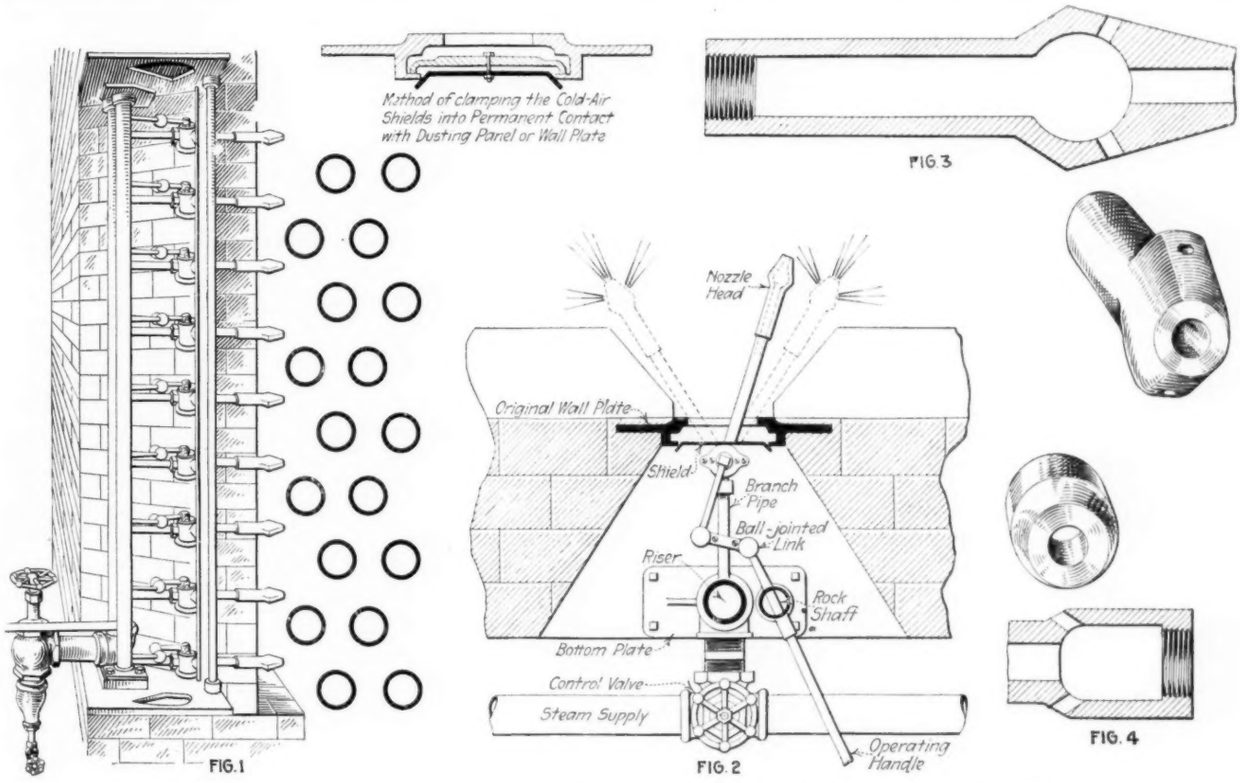
Bayer Soot-Blower System

A system for removing soot and fine ashes from the heating surfaces of vertically baffled water-tube boilers, and based essentially upon projecting powerful oscillating steam jets into, through and laterally across the spaces between the horizontal rows of tubes in planes parallel to the pitch of the tubes, is manufactured by the Bayer Steam Soot Blower Co., St. Louis, Mo.

The blower is so designed that the parts can be placed without changing or cutting brickwork and walls, the original cleaning openings and internal wall angles in the side of the boiler being ample to permit a wide swing of the nozzle heads.

The nozzles are divided into groups of individually controlled blowing units, Fig. 1, one of which serves each gas pass, or compartment, and is fitted into each of the cleaning pockets originally provided along the side wall for hand cleaning. The individual units receive steam in common from the superheater or from a 2-in. opening in the top of the main steam header, the steam-supply line passing below the blowing units and having a drain at the lowest point to permit the removal of any possible condensation just before each blowing operation.

Each blowing unit has a vertically and centrally located riser connected to the steam line and controlled by its own valve. Short branches extending inward from the riser in a vertical series, and equal in number to the space planes between the horizontal rows of tubes, connect the riser with swing joints, the moving members of which are attached through nipples to the three-way nozzle heads within the furnace wall. A vertical rock-shaft at the side of the riser, having a single operating handle, connects with the moving members of the swing



FIGS. 1 TO 4. SHOWING DETAILS OF THE BAYER OSCILLATING SOOT BLOWER

joints through links and levers. Using the operating handle and rocking the shaft alternately from the right to the left limits, the series of nozzle heads is correspondingly swung back and forth through wide arcs, Fig. 2, and if the steam has been turned on it will be delivered from each nozzle in three separate jets which sweep the gas pass, or compartment, and agitate and remove the soot and ash deposit.

The nozzle heads are designed according to their location. Fig. 3 shows the standard type used in the second and third passes to produce one main and two auxiliary sidewise jets in the plane of oscillation. Fig. 4 shows the standard first-pass type, which is made heavier than first- and second-pass type to better withstand the more intense heat of the first pass.

The type of swing joint used, in addition to permitting easy installation, inspection and overhauling, may be brought squarely up against the cover shield which closes the cleaning pocket and is tilted to the same pitch as the tubes. This permits the steam to sweep at all points within the plane spaces between the tubes, which is essential to thorough cleaning.

The swing joints all being bolted to a metal cover shield that fits against and is clamped to the two strips on the wall plate (Fig. 1), the cleaning pocket is made permanently air-tight and the unit is located when the shield is clamped into place. The riser and rockshaft terminals at the top and the bottom of the blower unit are special cast plates each having four heavy setscrews that fit against the iron lintel and sill of the cleaning pocket. Adjustment of the units in permanent position can be completed while the boiler is in service, and the rest of the installation is merely a matter of piping.

To avoid possible inefficiency in boilers having many vertical rows of tubes, the risers in the blower units are divided in the middle by a blind coupling, and separate steam supplies are piped to the top and bottom halves.

Double-deck boilers having two cleaning pockets, one above the other, for each pass are provided with a corresponding double arrangement of blower units, each having its own branch and controlling valve.

The units of this system either with or without modification can be applied to almost any other make of vertically baffled water-tube boiler.

The Action of Air in Surface Condensers

By Paul A. Bancel

SYNOPSIS—Decrease of air leakage with load on turbine due to higher steam pressure in lower stages of turbine. Active condensing surface varies with conditions of vacuum and water temperatures. Steam lanes not desirable. Tubes should be spaced evenly and on diminishing centers as they approach the outlet.

Referring to the article by C. F. Hirshfeld in the Feb. 29 issue, the writer wishes to point out the probable source of the variable part of the air leakage indicated in his Fig. 1. This chart showed that the air leakage decreased from 5 cu.ft. per min. at 2,000 kw. to 3 cu.ft. per min. at 8,000 kw.

If it is assumed that the leaks in the system, which are principally at the joints, act as orifices with which the downstream pressure has no effect upon the flow of air if it is less than about half the upstream pressure, then all of the condenser and the low-pressure end of the turbine where the vacuum is 15 in. or higher can be eliminated. As the load on the turbine is increased, the pressure rises throughout the turbine and in some region where there was formerly 15 in. of vacuum, there may now be 5 or 10 in. Similarly, in another part where there was formerly 5 in. of vacuum, there may now be several pounds' pressure above atmosphere. The air leakage will thus decrease. The leakage through joints subjected to 15 in. or more vacuum in the turbine and condenser will remain the same, but the sum of the two quantities will decrease according to some such curve as given in the chart previously mentioned.

Air inleakage may be reduced by sealing the joints, as shown in the various sketches of Fig. 1. The joints in the turbine casing, also between turbine and condenser, in the condenser and in the connection between the condenser and the air pump can all be sealed in the same manner.

Later on in the article, Mr. Hirshfeld points out that the work of a condenser may be divided into two parts—that of active condensation and that of cooling the residue to decrease the work of the air pump. It is stated that "most of the surface is used for work of the first kind, only a small fraction of the surface being used for cooling."

If steam containing little air has access to clean condensing surface, the rate of condensation will be high, or in terms of heat transmission, the coefficient per degree difference per square foot of surface will be high. Tests show that coefficients of 1,000 to 1,500 are obtained under these conditions. On the other hand, the process of chilling the air-stream residue is inefficient, the coefficient of heat transmission being small. Tests by Professor Josse showed that in the top of a condenser where active condensation was going on, the coefficient of heat transmission was 990, whereas in the bottom section, where there was little condensation but where the surface was employed in air chilling and concentration, the coefficient was too small to be calculated. As a result the coefficient of heat transmission based on all the surface in the condenser was only 360, which, contrasted to the figure of 990, indicated approximately the proportion of actively working surface.

If it is assumed that the "small fraction" mentioned by Mr. Hirshfeld is one-tenth of the surface, then the average coefficient of heat transmission for all of the condenser should be, roughly, nine-tenths of that in the zone of active condensation or, say, $\frac{9}{10} \times 1{,}000 = 900$.

This would be the coefficient obtained if only that surface set aside in some designs for the internal air cooler were employed in air cooling and all the rest of the surface were active in condensing steam.

However, this does not occur, since high coefficients of heat transfer are rarely obtained in actual practice, even with new condensers. In large high-vacuum units coefficients of 300 to 400 are considered good under summer conditions when the relatively high mean-water temperature is slightly in favor of the performance and the more moderate vacuum is much in favor of the performance, while coefficients of 100 to 200 are obtained in the winter time with cold water and maximum vacuum.

Better efficiency of surface at high vacuum can be obtained by preventing air inleakage, increasing pump capacity and decreasing the pressure drop or pneumatic resistance of the condenser by proper arrangement of steam path and spacing tubes, these being of great importance at high vacuums.

In purchasing high-vacuum condensers, comparison should be made of the pneumatic resistance of the structure on the basis of velocity of flow at each row of tubes and the number of rows in series through which steam must flow. Attention should also be given to possibilities for the formation of air pockets out of the line of flow, considering both transverse and longitudinal sections.

By analyzing the temperature depression in a given condenser into that due to pressure drop and that due to partial air pressure, it is possible to determine whether flow conditions or air conditions offer the greater possibility for improving efficiency and vacuum.

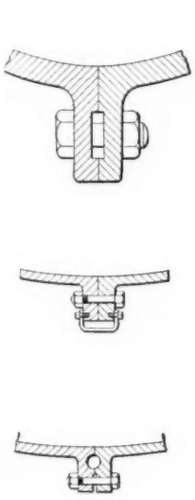


FIG. 1

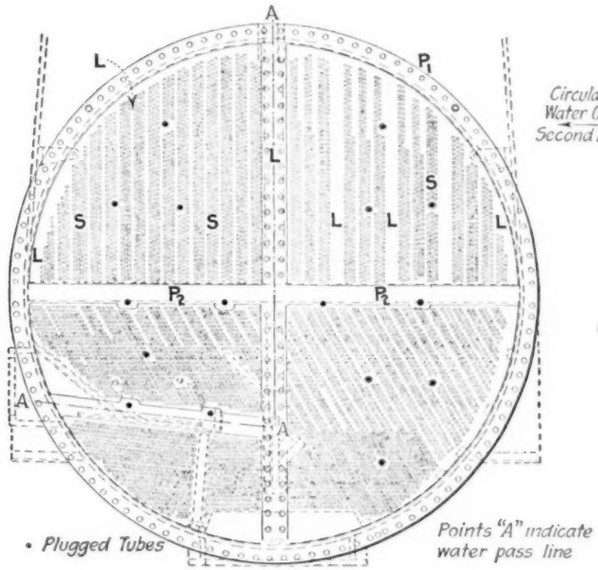


FIG. 2

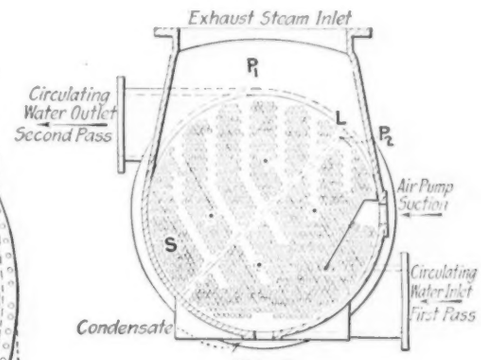


FIG. 3

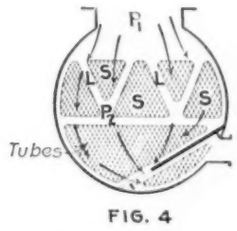


FIG. 4

FIGS. 1 TO 4. DETAILS OF CONDENSER CONSTRUCTION, SHOWING WHERE PROBABLE LOSSES OCCUR

These results indicate that the proportion of surface actively condensing steam is somewhat smaller than that to be inferred from Mr. Hirshfeld's article, and that the proportion varies with conditions of vacuum and mean-water temperature, being more in summer with moderate vacuum and warm water and less in winter with cold water and higher vacuum.

High coefficients of heat transfer are found to coincide with: (1) Heavily loaded condensers and moderate vacuum; (2) summer conditions when the vacuum is moderate and water temperature high; (3) small condensers in which the hydrodynamic resistance of the shallow tube bank is low; (4) large air-pump capacity, either in the form of a large displacement pump, steam augmenter or centrifugal water-jet pump consuming considerable power.

The influence of the various factors on surface-condenser performance are given in a paper by the writer and George H. Gibson in the *Journal* of the American Society of Mechanical Engineers for January and February, 1916. Some of the conclusions will serve to summarize the foregoing discussion and are repeated here:

The lower average heat-transfer coefficient obtained with cold circulating water is due principally to a greater proportion of the surface being inactive, which in turn is due to greater pressure drop incidental to the flow of a larger volume of steam and to the greater proportion of air to vapor existing in the mixture stagnating in the outlet zone of the condenser.

A high velocity of circulating water, or the equivalent increase in the water-film agitation by the use of cores or spirals, is desirable in the tubes of the active zone of a condenser; and this can be obtained without additional power consumption in pumping the circulating water, by reducing the velocity of the water through the tubes of the inactive zone.

The argument has been advanced that condensers of recent manufacture are designed so that the surface is in full activity throughout the shell, as proved by the fact that the rise in temperature of the circulating water flowing through the tubes in the bottom pass of a condenser is as great or greater than that in the top pass. Unfortunately these condensers do not give high coefficients of heat transfer, indicating that a large part of the surface is still inactive. It is found that the condition of equal or greater work in the lower pass coincides with designs in which openings or steam lanes have been provided to pass the steam down to the lower tubes, thus allowing the steam to bypass groups of tubes in the top of the condenser. This is a source of a twofold loss: First, the amount of active condensing surface is reduced, representing a waste of investment in equipment, and secondly, condensing water is uselessly pumped through inactive tubes, so that the rise in temperature of the water passing through tubes upon which condensation can occur is greater than it should be. Only part of the condensing water does useful work. Thus if the temperature rise of the water, as measured by ther-

mometers in the boxes at the ends of the condenser, were 6 deg., what might be actually occurring is a rise of 12 deg. on half of the water and no rise on the other half. This, however, is masked by the fact that the average temperature of the mixture of the discharge from various tubes is read by the thermometer in the water box or outlet pipe.

It is interesting to notice how we work in circles; only a few years ago the great dread of the condenser designer was a short-circuit of steam to the air pump. Particular care was exercised in designing the shell and in the layout of tubes, to prevent any free path for flow of steam down into the condenser.

With the demand for higher vacuums another difficulty arose. It was found that the difference of vacuum between the top and bottom of a condenser was quite considerable and the great bugaboo became "pressure drop." Mercury-column readings showed a vacuum at the top of the condenser of, say, 28½ in. and 29 in. at the air pump. It was obvious that if the vacuum at the air pump could only be maintained at the turbine exhaust, thousands of dollars would be saved annually.

The solution advanced and applied in many instances of large and expensive equipments was to open up steam lanes or passages which, it was reasoned, would let the steam down into the condenser and thus reduce the pressure drop. The old fear of a short-circuit to the air pump was forgotten.

Three such designs are reproduced in Figs. 2, 3 and 4. The tubes S at the center are more or less short-circuited in each case by the lanes or steam passages L, which tend to equalize the pressures P_1 and P_2, thus reducing the pressure drop, but unfortunately without increase in vacuum at the turbine exhaust.

The engineer can point with pride to a drop of, say, 0.1 in. as against 0.5 in. formerly, overlooking that this had been accomplished by reducing the 29-in. air-pump vacuum to 28.6 in. without change in the vacuum of 28½ in. at the turbine.

The correct solution is to space out the tubes evenly all the way across the path of the steam, leaving no preferential lanes down which the steam can short-circuit. This spacing should be graduated, starting out with tubes on extra-wide centers and following with tubes on diminishing centers.

Don't Lose Your Pep!

By Alfred W. Conklin

The following was suggested after reading your editorial on "The Temptation To Let Up," which appeared in the April 18, issue:

Don't lose your pep! Set out to win
  Promotion and the highest place
Your job affords. Dig in!
  Success must come. So keep the pace
You started when the work was young.

Don't lose your pep! Nor yet give in
  To worries and the many doubts
That will assail. Just grin!
  Success will come. And manly bouts
Increase your strength for tasks undone.

Don't lose your pep! Nor falter in
  Your stride. You've won a place
That brings reward. Pitch in!
  Success has come. Still keep the pace
You started when the work was young.

Diagram for Computing Flow of Steam in Pipes

By Howard Harding

The accompanying diagram is a graphical development by the author of the well-known formula for the flow of steam in pipes:

$$W = 87 \sqrt{\frac{D\,(p_1 - p_2)\,d^5}{L\left(1 + \frac{3.6}{d}\right)}}$$

where

W = Weight of steam flowing, in pounds per minute;
D = Density in pounds per cubic foot of steam at the mean absolute pressure;
p_1 = Absolute pressure in pounds per square inch of the steam at the entrance section of the pipe;
p_2 = Absolute pressure at the exit section;
d = Diameter of the pipe in inches;
L = Length of the pipe in feet.

It will be noticed that the diagram is so arranged that all the variables are taken into account and no slide rule or other numerical computation is necessary. Given any four of the following factors, the remaining one may be determined: (1) Pounds of steam per minute; (2) average pressure in pounds absolute; (3) pressure drop; (4) pipe size; (5) length in feet. For the solution of many problems the directness of the diagram will be found of advantage.

It may be interesting to outline the procedure in solving different problems.

Given the weight of steam per minute, the average pressure, the pressure drop and the pipe size, to determine the length in feet: Start at the intersection of the abscissa representing pounds of steam per minute and the ordinate representing the average pressure; thence proceed upwardly parallel to the oblique guide lines on the left side of the diagram until "reference line for average pressure" is intersected; thence horizontally to the vertical line representing the given pressure drop; thence obliquely downward to "reference line for pressure drop"; thence horizontally to intersect the inclined line representing the given pipe size. The vertical line passing through the last intersection will determine the required length of pipe in feet.

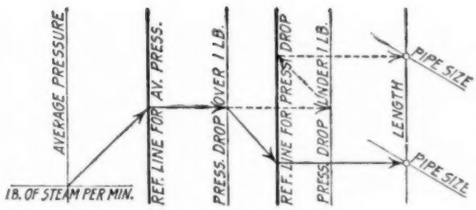
Given steam per minute, average pressure, pressure drop and length of pipe, to determine the pipe size: The procedure is the same as for the first problem except that the last intersection is with the given length of pipe. The oblique line through that intersection determines the required pipe size.

To determine the weight of steam per minute or the average pressure: Start at the intersection of the given pipe size and the given length and proceed horizontally left to intersect the "reference line for pressure drop"; thence obliquely upward to intersect the line representing the given pressure drop; thence horizontally to the "reference line for average pressure"; and finally obliquely downward to intersect either the given steam per minute or the given average pressure. The intersection will automatically determine the required average pressure or the weight of steam per minute.

To determine pressure drop: Starting at the intersection of the given steam per minute and the given average

Given any four of the following conditions, the remaining one may be determined.

1 - Steam per Minute (Lb.)
2 - Average Press. Abs. = (Initial Press - ½ Press. Drop)
3 - Pressure Drop
4 - Pipe Size
5 - Length in Feet



FORMULA: $W = 87\sqrt{\frac{D(P_1 - P_2)d^5}{L(1 + \frac{3.6}{d})}}$

TO DETERMINE POUNDS (WEIGHT) OF STEAM PER MIN. OR AVERAGE ABSOLUTE PRESS.

Starting at intersection of given pipe size and length, proceed along the arrows in a reverse direction.

(Dotted lines show procedure when pressure drop is less than one pound).

TO DETERMINE PRESSURE DROP, LB. PER SQ. IN.

Starting at intersection of steam per minute and average pressure, proceed along arrows to the right. At the same time starting from intersection of pipe size and length, proceed along arrows to the left. Intersection of these courses will give the pressure drop.

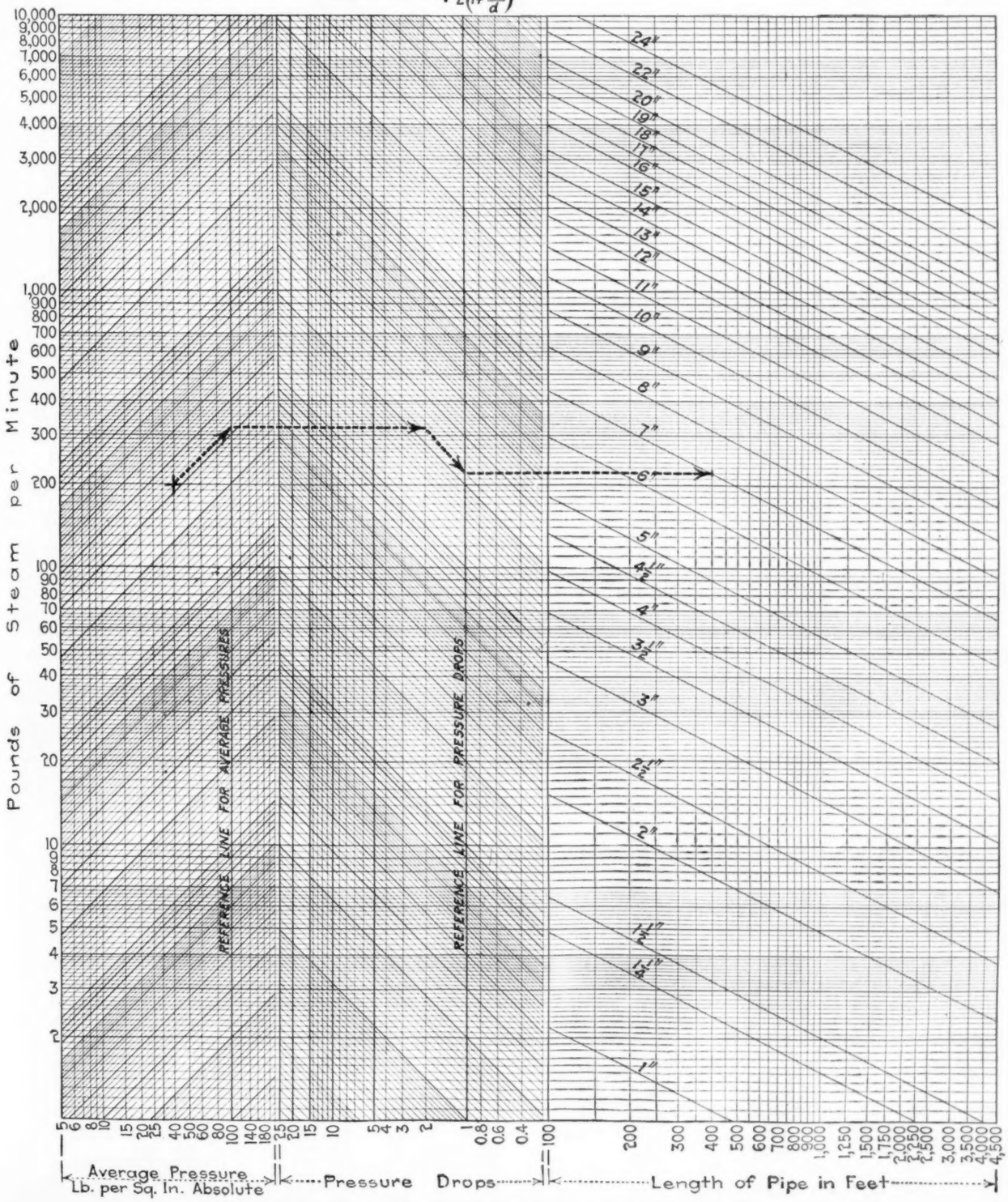


DIAGRAM FOR COMPUTING THE FLOW OF STEAM IN PIPES

pressure, proceed to the right as in the first problem. At the same time starting at the intersection of the given pipe size and the given length, proceed to the left as outlined in the last problem. The diagonal that is intersected at the reference line, by its intersection with the horizontal first traced, will determine the drop.

The solution of the following problem is indicated by connected heavy dotted lines on the diagram: What size of pipe will be required to transmit 200 lb. of steam per minute a distance of 400 ft. at an average pressure of 40 lb. absolute with a pressure drop of 2 lb.? As indicated on the diagram, a 7-in. pipe will be required.

Testing Safety Valves at the Naval Engineering Experiment Station

By Lieut. J. L. Kauffman*

SYNOPSIS—Extracts from the safety valve specifications of the United States Navy and how such valves are tested before acceptance.

Marine safety valves are tested at the United States Naval Engineering Experiment Station to determine their suitability for use in the naval service. Some tests of locomotive safety valves have been made for the Interstate Commerce Commission. The tests at this station are required under the present specifications of the Navy Department, from which the following are extracts:

Two boiler safety valves and such other safety valves as the Bureau of Steam Engineering may direct shall be selected at random by the inspector of machinery and forwarded to the Engineering Experiment Station, Annapolis, Md., for test.

The valve, while discharging, must be quiet and free from chattering.

Simmering will be allowed at 2 lb. before popping, but the valve closure must be sharp.

The design of the huddling device must be such that the adjustment while being varied must remain symmetrical with the axis of the valve.

The valves must be adjustable from a 5-lb. to a 10-lb. blowdown without disassembling the valve and without causing chattering.

The springs will be of material round or square in cross-section and of such a length and diameter that the fiber stress will not exceed 55,000 lb. per sq.in. when the spring is compressed to the position it would assume were the valve lifted so as to give its rated discharge.

They will have spherical bearings at their ends, or they will be connected to the compression plates in such a manner as to insure a proper distribution of pressure.

The fiber stress of springs will be calculated from the torsion of the wire and the direct shearing action of the load by one of the following formulas:

$$\text{For elliptical section: } S = \frac{16\ Q \cos\theta\ R}{\pi\ mn^2} + \frac{Q \cos\theta}{A}$$

$$\text{For rectangular section: } S = \frac{9\ Q \cos\theta\ R}{2\ bh^2} + \frac{Q \cos\theta}{A}$$

In which

S = Maximum unit shearing stress in pounds per sq.in.;
Q = Load on spring in pounds;
R = Mean radius of coil measured from axis of spring to center of gravity of section in inches;
θ = Angle of inclination of coil to plane perpendicular to axis of coil, and for approximate calculations $\cos\theta$ may be taken as unity;
A = Area of section of coil in square inches;
π = 3.1416;
m = Axis of elliptical section measured parallel to axis of spring in inches;
n = Axis of elliptical section measured perpendicular to axis of spring in inches;
b = Length of base of rectangular section measured parallel to axis of spring in inches;
h = Altitude of rectangular section measured perpendicular to axis of spring in inches.

*Lieutenant, junior grade, United States Navy.

For circular sections m = n = diameter of section; for square sections b = h, length of side; for trapezoidal sections b = average length of the two bases.

The angle of the valve seat will be as near 0 deg. as practicable and not exceed 45 deg. from the horizontal.

The valves when tested by steam must lift $\frac{1}{10}$ in., with a corresponding blowdown not to exceed 10 lb. to fulfill the requirements of the specifications.

The springs of all safety and relief valves will be tested to insure the satisfaction of the requirements as to load.

The springs of all valves will be exposed to view.

All valves are tested on a marine-type oil-burning Mosher boiler having 4,537 sq.ft. of heating surface and fitted with 11 burners. The steam drum is 48 in. diameter and 14 ft. long. A 6-in. gate valve is mounted on the boiler drum with the safety valve clamped to the gate valve, so that the inlet of the safety valve is 1 ft. from the opening in the drum. The exhaust is either to the atmosphere through a 6- to 12-in. pipe or through a long connection to a surface condenser.

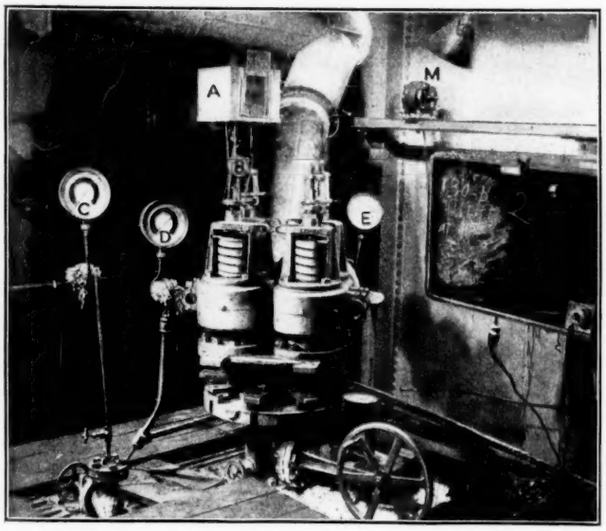


FIG. 1. VALVES READY FOR A TEST

Fig. 1 shows a safety valve in place with the testing apparatus installed. There are shown two duplex valves on a Y-base with three units gagged and the fourth ready for test; *A* is the recording apparatus, *B* is the micrometer lift gage, *M* the motor for driving the recording apparatus, *C* a test gage connected directly to the boiler drum, *D* a test gage connected above the gate valve and *E* a test gage connected to the piezometer ring on the exhaust line to indicate the back pressure. Under the valve on the right side of the gate valve can be seen the calorimeter with the attachments for

determining calorimeter pressure and temperature. At the extreme right are two electric switches used to operate signal lights placed near the front of the boiler to indicate to the fireman the number of burners required

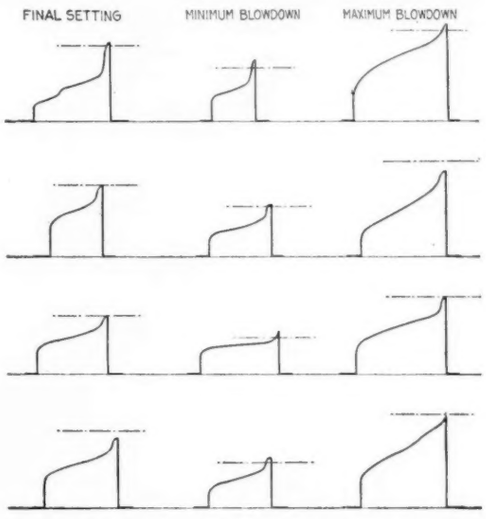


FIG. 2. DIAGRAMS OF LIFT OF SAFETY VALVES

to get close adjustment of blowdown. By this method the steam pressure can be carefully regulated, one burner being ordinarily used on blowdown determinations that the action may be sufficiently slow to obtain accurate readings.

The valve is first popped several times to heat the body uniformly. The blowdown and lift are then recorded for five or six pops, and the required adjustments made to determine whether or not the valve will give the minimum and maximum blowdowns specified. When the maximum and minimum blowdowns are obtained, the sustained lift is determined by lighting a sufficient number of burners to keep the valve blowing and then reading the lift for the range of pressures, including the popping pressure. For instance, if the valve pops at 260 lb. and seats at 255 lb., the lift is read with the valve blowing at 256, 258, 260, 262, 260, 258, 256 and 255 lb. gage. Curves are plotted between steam pressure and lift, to determine the lift at the popping pressure. The recording apparatus used gives a graphic representation of the lift when the valve is opened and closed. As this apparatus is carefully calibrated, the lift gage readings can be checked by measuring the diagrams taken. The diagrams show the action of the valve and spring from the instant of popping until the valve closes, indicating whether the valve hangs during the closing. In Fig. 2 are several typical lift diagrams. The paper moves from left to right so that the right-hand end shows the lift at popping and the left-hand end the lift at closing. The dotted lines are drawn in to show the values obtained on the sustained lift for the same popping pressure and blowdown.

Careful observations are made for leakage and simmering; a feather of steam continuing to issue from the exhaust after the valve has been closed several minutes is considered leakage.

To obtain the actual discharge and the discharge factor for a given valve lift and steam pressure based on the theoretical discharge, a capacity run is made. The lift at a blowing pressure corresponding to the rated popping pressure is maintained constant by regulating the number of burners in use. Conditions becoming constant, a two-hour run is made. The valve-lift gage is read every 15 sec., the steam pressure every minute, while the feed-tank level, boiler-drum water level, calorimeter temperature and pressure and weight of water discharged by the calorimeter are noted every 5 min. Popping pressures and blowdown observations are made just before and after each capacity run. After the capacity run a leakage run is also made on the boiler. Everything on the boiler remaining as on the capacity run and with the safety valve and feed lines closed, the steam pressure is kept constant and the drum-water level noted at 1-min. intervals for 2 hr. The drop in level in the drum is a measure of the leakage. The different items used in determining the actual discharge, the discharge factor and their derivations, are as follows:

1. Date	For reference
2. Valve number	For reference
3. Stem number	For reference
4. Duration of run, hr.	For reference
5. Room temperature, deg. F.	For reference
6. Barometer, in. of mercury.	From data, corrected for temperature according to Smithsonian Meteorological Tables
7. Blowing pressure, lb. per sq.in. gage	Observed
8. Blowing pressure, lb. per sq.in. abs.	Item 7 + 0.491 × item 6
9. Calorimeter pressure, lb. per sq.in. abs.	0.491 × item 6 + calorimeter gage pressure

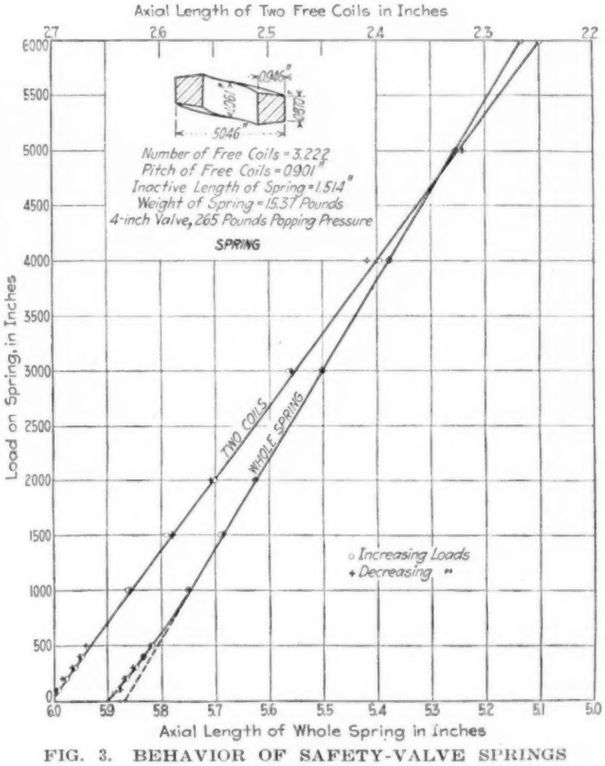

Axial Length of Two Free Coils in Inches
Load on Spring, in Inches
Axial Length of Whole Spring in Inches
Number of Free Coils = 3.222
Pitch of Free Coils = 0.901"
Inactive Length of Spring = 1.514"
Weight of Spring = 15.37 Pounds
4-inch Valve, 265 Pounds Popping Pressure
SPRING
TWO COILS
WHOLE SPRING
Increasing Loads
Decreasing "

FIG. 3. BEHAVIOR OF SAFETY-VALVE SPRINGS UNDER TEST

10. Calorimeter temperature, deg. F.	Observed
11. Valve lift, in.	Observed
12. Contact diameter, in.	As measured
13. Water fed to boilers, lb.	As weighed
14. Water lost by leakage, lb.	From leakage test
15. Water loss through calorimeter, lb.	As weighed
16. Water correction for difference in drum level, lb.	From difference in level in calibrated boiler drum

17. Steam-dryness fraction.... From formula

$$x = \frac{He + Cp(Te - Ts) - qb}{rb}$$

where

He = Total heat of steam at calorimeter pressure, item 9;
te = Calorimeter temperature, item 10;
ts = Temperature corresponding to calorimeter pressure, item 9;
cp = Specific heat of superheated steam corresponding to calorimeter pressure and superheat;
qb = Heat of liquid at boiler pressure, item 8;
rb = Heat of vaporization at boiler pressure, item 8.

18. Steam-moisture fraction... From formula

$$y = 1 - x$$

where

y = Moisture fraction of steam;
x = Dryness fraction of steam.

19. Valve-opening area, sq.in... From formula for valves with seats of 45 deg.:

$$a = 1.111\ b\ (2d + b)$$

Valves with flat seats:

$$a = b \times c$$

where

a = Discharge area in square inches (item 19);
b = Lift of valve in inches (item 11);
c = Circumference of seat;
d = Contact diameter of valves in inches (item 12).

20. Theoretical discharge, lb. per hr.....From formula

$$W = 3{,}600\,\frac{ap^{0.97}}{61}\,(1 + y)$$

where

W = Theoretical discharge in lb. per hr. (item 20);
a = Discharge area in square inches (item 19);
p = Absolute blowing pressure in pounds per square inch (item 8);
y = Moisture fraction of steam (item 18).

21. Actual discharge, lb. per hr.

$$\frac{\text{Item 13} - (\text{item 14} + \text{item 15}) \pm \text{item 16}}{\text{Item 4}}$$

22. Discharge factor $\left(\frac{\text{actual}}{\text{theoretical}}\right)$.. $100\,\frac{(\text{item 21})}{(\text{item 20})}$

After the tests under steam, the valves are removed and all parts carefully examined for wear. Each spring is then compressed in a testing machine. The springs are maintained in a central position by two cast-iron disks with spherically shaped ends, a lug on each disk fitting into the hole in the spring. The length of the spring is measured for various loads from zero to the maximum and back to zero again. The axial length of the free coils is measured in the same manner.

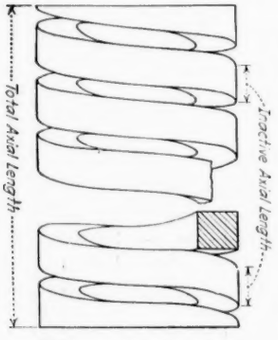


FIG. 4. ACTIVE AND INACTIVE PARTS OF SPRING

The values obtained are plotted in the curves shown in Fig. 3. As may be seen from the curves, the points for increasing and decreasing loads generally fall practically on the same line, showing no permanent set in the springs under normal conditions. The deviation from a straight line at the lower end of the curve is due to the formation of the ends of the spring.

The maximum fiber stress occurs at the middle of the inside edge of the section of spring coil. The maximum stress is made up of four component stresses, respectively, due to torsion, to direct shear, to direct compression, and to bending. The last two components affect the total stress very little and are therefore neglected. The first two components produce the shearing stress which is allowed by the specifications. The maximum fiber stress is calculated by the formula in the specifications quoted.

The load Q is selected to correspond to the axial length of the spring when the valve is lifted to give its rated discharge. The axial length of the spring when the valve is closed is determined by the aid of the curves in Fig. 3 to correspond to a load equal to the rated popping pressure acting upon the area within the circle of contact between the valve disk and seat, the diameter of which is denoted by d.

The axial deflection produced by the load is also due to torsion, to direct shear, to direct compression and to bending of the bar forming the coil; and, as before, the latter two are considered negligible.

The relation between the axial deflection and the load on the spring is given by

$$\delta = \frac{Q\cos^2\theta R^2 L}{CJG} + \frac{Q\cos^2\theta L}{AG}$$

in which

δ = Axial deflection per coil in inches corresponding to

Q = Load on spring in pounds;

R = Mean radius of coil in inches measured from axis of spring to center of gravity of section;

$L = \sqrt{(p - \delta)^2 + (2\pi R)^2}$ true length in inches of one free coil actively opposing the compression of the spring, where

p = Axial pitch in inches by measurement of the free coils at no load;

$\cos\theta = \frac{2\pi R}{L}$;

C = 0.841, St. Venant's constant for the resistance to torsion of bars of nearly square section;

$A = h \times \frac{B + b}{2}$ area of trapezoidal section in sq. in., where

h = Altitude of trapezoidal section in inches;

b = Smaller base in inches;

B = Larger base in inches;

$J = \frac{h}{48}(B^3 + B^2b + Bb^2 + b^3) + \frac{h}{36}\frac{(B^2 + 4Bb + b^2)}{B + b}$ = polar moment of inertia of trapezoidal section in inches^4;

$J = \frac{\pi mn}{64}(m^2 + n^2)$ = polar moment of inertia of circular section in inches^4, where m and n same as given in specifications;

G = Shearing modulus of elasticity in pounds per square inch.

The numerical value of the shearing modulus of elasticity is obtained by substitution in the foregoing formula.

Owing to the formation of the spring to secure a solid bearing, the number of coils available for actively resisting the load is less than the total number of coils, so that at each end there is an inactive length of spring. The number of equivalent free coils and the length of the inactive part cannot be estimated accurately by observation, but may be calculated from the curves in Fig. 3. For example, the axial deflection of the spring (Fig. 3) under a load of 5,000 lb. is 5.868 — 5.259 = 0.609 in., while the deflection of the two free coils under the same load is 2.703 — 2.325 = 0.378 in., or for one free coil it is $\frac{0.378}{2}$ = 0.189 in. Therefore in the spring

there are actually $\frac{0.609}{0.189} = 3.222$ free coils. At very light loads the number of free coils is increased as shown by the curvature of the lower end of the line of the drawing, the value 5.868 being taken at the prolongation of the straight part. To obtain the inactive length, for example, at 2,500 lb. we have the active length $3.22 \times \frac{2.514}{2} = 4.050$, where 2.514 is the length of two free coils, taken from the curve. As the axial length of the

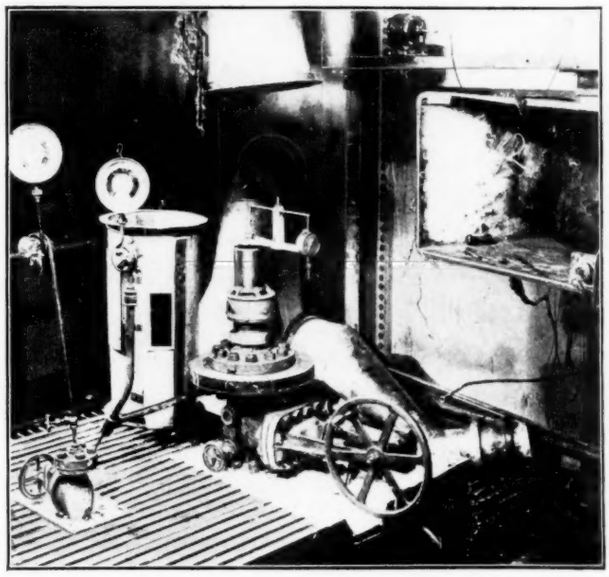


FIG. 5. SHOWING THE LIFT APPARATUS IN THE CENTER OF PICTURE

whole spring at this load is 5.564, the inactive length is 5.564 — 4.050 = 1.514 in. Practically the same values for the inactive length will be found at all loads except the very lightest.

Fig. 4 is not drawn to scale, but simply indicates what is meant by free coils and inactive part.

The stiffness of the spring is measured by the load per inch of axial deflection, the less flexible spring requiring the greater load. These values are determined from the curves in Fig. 3.

Tests have been made on springs by compressing them solid, allowing them to remain so for several hours and then determining the set, if any, as the loads are decreased.

In the tests on safety valves for locomotive and stationary boilers the determination of the range of blowdown is usually omitted, only the minimum blowdown being obtained. The principal object of these tests is to determine the capacity, which is done in the same manner as that described in a previous paragraph. A different arrangement is required, however, in that the valve must be covered by a hood to carry off the steam and a different lift-gage apparatus used. This lift apparatus is shown in Fig. 5, where the hood has been removed and is standing to the left of the valve. The narrow opening in the hood is made for the micrometer lift-gage mechanism to project through.

In special tests arrangements are made to give high back pressures by closing a valve placed in the exhaust line. This has been done on tests of stationary-boiler valves (for the Panama Canal Commission) to determine their ability to function properly when they have exceptionally long exhaust lines.

In all tests of marine valves for use in the naval service, the practical features are always kept in mind. None of these tests is considered too theoretical for practical purposes, as the highest quality in all parts is a necessity, and it is imperative that all possible points be covered fully in the tests. The fact that a safety valve must function regularly with the smallest possible leakage of steam when the valve is closed is always kept in mind; also, that a valve is not a satisfactory one if it requires extremely delicate adjustment or if it requires constant overhauling.

The formulas used in determining the various values required are the work of W. L. De Baufre, a member of the technical staff at the experiment station and are considered to meet the requirements from both a theoretical and a practical standpoint.

The National Boiler Girth Seam Protector

The National Boiler Protector Co., 740 Reibold Building, Dayton, Ohio, has recently brought out the device shown in the accompanying illustration to protect the girth seam of horizontal return-tubular boilers located over the fire. Owing to overheating of the metal where the plates overlap and sudden cooling by the opening of the fire-doors, this particular seam has given trouble, the expansion and sudden contraction in time developing fire cracks that may become serious if not given immediate attention.

The protector guards against any sudden change of temperature at this point. It is made up of five sections to cover about 50 in. of the lower part of the seam. Each section is independent of the others, so that there is no interference from expansion or contraction.

A nut wedged between two rivet heads and tapped for a ½- or ⅝-in. tap screw supports a metal lug, which in

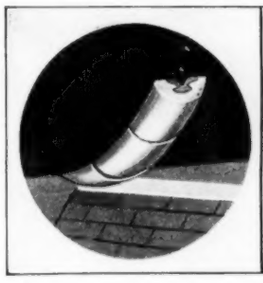


FIG. 1. APPLICATION TO GIRTH SEAM

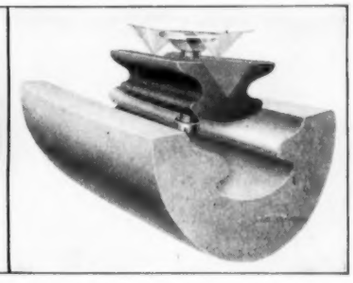


FIG. 2. SUPPORT AND PROTECTING TILE

turn supports the firebrick tile protecting the supporting member and the seam. In the illustration it will be seen that the lug has a V-notch on the inner side so that it may fit down over the rivet heads and come in close contact with the plate. On the exterior it is paneled to receive the firebrick tile, which slips over the lug and also fits up tight against the metal of the boiler, covering the rivet heads, but leaving a sufficient space at the rear end so that the calking edge may be observed by the operator.

The protector is guaranteed for one year, and it is claimed that it has been giving excellent results in service.

The Plant Crew Reduces Ash-Handling Costs

By A. A. Norrman

SYNOPSIS—The crew of the plant of the Eastern Pennsylvania Light, Heat and Power Co. built an ash-conveying system that saves nearly $12 a day in labor alone.

In 1907 the new boiler plant of the Eastern Pennsylvania Light, Heat and Power Co., Pottsville, Penn., consisted of six 320-hp. water-tube boilers, furnishing steam at 160 lb. In 1909 two more 320-hp. boilers were added, and in 1913 two more were installed.

The boilers were equipped with shaking grates, with about 30 per cent. air space, for burning No. 1 buckwheat coal, but shortly after the plant was started, the fuel was changed to buckwheat No. 2, or rice coal.

The ashes were removed by hand, requiring three men by day and three men by night, wheeling it to a pile outside the boiler house. The operating cost of the boiler room was high, and to reduce it, the following changes were made: Dumping grates with 14 per cent. air space were selected to burn the lower grades of anthracite fuel.

FIG. 1. VIEW IN ONE OF THE ASH TUNNELS

The kind now used is No. 3 buckwheat, or barley, coal, containing about 11,000 B.t.u. per lb. and costing 30c. less per ton than rice. The brickwork on top of the boilers was removed, and the tops were covered with 2½ in. of high-temperature insulation. The bricks were relaid on top of the covering.

All cracks and leaks in the boiler settings were plugged with a mixture of one part of cement, two parts fine sand and eight parts asbestos. The opening in the first pass above the front baffle was reduced from 36 in. to 27 in., effecting a reduction in the flue-gas temperature. The individual boiler dampers were provided with chains running on pulleys to the front of the boilers, so the firemen can maintain as nearly as possible a balanced draft. Air for the forced draft is supplied by an engine-driven fan. Indicating steam-flow meters were installed, one on each

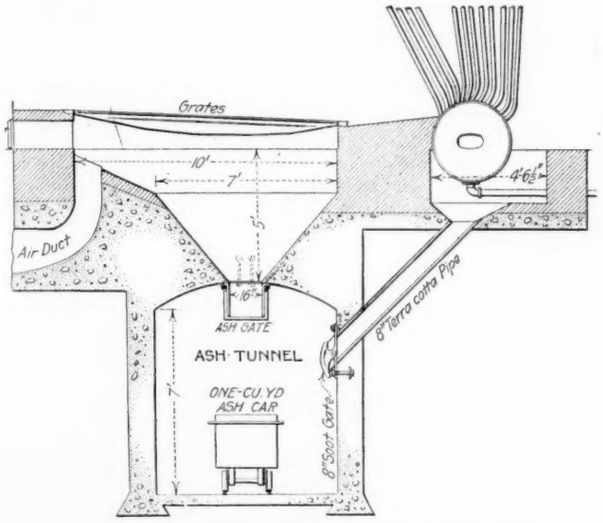


FIG. 2. SECTION OF ASHPIT AND TUNNEL

boiler, to show the steam output of the boilers. Pyrometers indicate the temperature of the flue gases for each boiler.

These changes effected a saving in coal consumption of 1 lb. per kw.-hr. in addition to 30c. per ton. The consumption of coal is about 80 long tons in 24 hr. The boiler-room expenses were still high, owing to the cost of

FIG. 3. SKIP-HOIST BUCKET

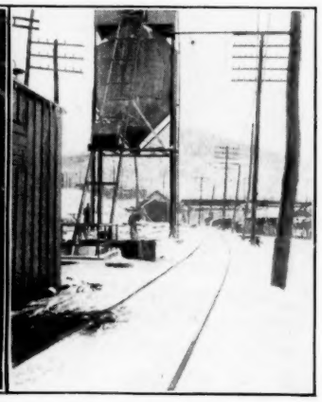


FIG. 4. THE ASH TANK

ash handling being $12 per 24 hr., not including hauling to the dump.

It was found possible to construct a tunnel under the boilers for taking the ashes from the ashpits by gravity, instead of pulling them by hand to the boiler-room floor. There are six boilers in one room and four in another. A tunnel 7 ft. wide by 8 ft. high was constructed under both boiler rooms. The length of the tunnels including the connecting tunnel is about 300 ft.

Fig. 1 shows a view of the large tunnel under the six boilers. As shown in Fig. 2, each ashpit under the boilers has a 16-in. cast-iron gate, used for dumping the ashes into a car of 1 cu.yd. capacity. The car is pushed by hand on an 18-in. gage track to the extreme end of the tunnel, where the ashes are dumped into a skip-hoist car, Fig. 3, which is electrically operated by an automatic reversing motor and dumps into a 300-cu.yd. steel bunker built over standard-gage track.

A view of the bunker is shown in Fig. 4. The present labor cost for removing the ashes from the pits to the bunker is 80c. per day; compared with former methods, the saving per day is $11.20 for labor alone.

With the exception of the skip hoist and bunker, which was built and erected by an outside firm to conform with the remodeling contemplated by the company, all changes and improvements in the boiler-plant operation and also the construction of the ash tunnel were made by W. P. Supplee, chief engineer of power stations, and the work was supervised by the writer, ably assisted by all those connected with the power plant. Besides reducing the labor, there is now no ash dust floating about the plant.

Calculating the Quantity of Ammonia Needed

By Charles H. Bromley

SYNOPSIS—Explains the factors that influence the quantity of ammonia needed in the refrigerating system and shows how to calculate the amount required.

To make the situation interesting assume that, the next time you are examined for a license, the examiner should pop this question: "Explain what the quantity of ammonia should be in the refrigeration system, and show how to calculate how much of the refrigerant should be circulated per hour for, say, a 50-ton plant carrying a back pressure of 25 lb. and a condenser pressure of 175 lb."

There must be enough ammonia in the system to insure a solid stream of liquid going through the expansion valve, and there should be enough gas evaporated from the liquid to make the evaporating coils take up heat equal to the capacity of the plant. In addition there must be enough surplus to allow for loss by leakage and for possible pocketing of the liquid somewhere in the system. It is evident that one of the items needed is the volume of the evaporating coils.

This is easily found either by calculating or by the use of a table of pipe dimensions found in engineers' books and catalogs. Table 1 gives these dimensions for pipe of evaporating-coil sizes:

TABLE 1. PIPE DIMENSIONS FOR EVAPORATING COILS

Internal Diameter, In.	Length per Cu.Ft.	Cu.Ft. per 100-Ft. Length
¾	270.25	0.370
1	166.9	0.599
1¼	96.25	1.038
1½	70.66	1.415
2	42.91	2.360

Suppose the 50-ton plant has 12,000 ft. of 2-in. pipe in the expansion coils. We have 120 × 2.360 = 283.2 cu.ft. of coil volume to be filled with gas at 25 lb. pressure. Now we need a table of properties of saturated ammonia to find the weight of a cubic foot of gas at 25 lb. Table 2 is from "Properties of Steam and Ammonia," by Goodenough.[1]

Referring to the table, we find that at 25 lb. gage, 40 lb. abs. pressure, ammonia weighs 0.1430 lb. per cu.ft. Therefore the weight of gas in the 12,000 ft. of 2-in. coils is 283.2 × 0.1430 = 40.6 lb.

[1]Published by John Wiley & Sons, New York City.

This quantity will do very well for the expansion coils, but ammonia also is needed for the condenser, receiver oil trap, etc. Practice has shown that the following quantities, mentioned by Luhr in his "Mechanical and Refrigerating Engineers' Handbook," if supplied in addition to the quantity necessary for the expansion coils, give amounts sufficient for the systems:

TABLE 2. PROPERTIES OF SATURATED AMMONIA

(Goodenough and Mosher; John Wiley & Sons, New York)

Pressure, Lb.	Temp., Deg. F.	Volume, Cu.Ft. per Lb.	Weight, Lb. per Cu.Ft.	Heat Content in B.t.u. of Liquid	Heat Content in B.t.u. of Vapor	Latent Heat in B.t.u. of Vaporization	Latent Heat in B.t.u. Internal
20	—15.9	13.45	0.0744	—50	534.0	584.3	534.7
21	—14.0	12.82	0.0780	—48.4	534.6	582.9	533.1
22	—12.2	12.27	0.0815	—46.5	535.1	581.5	531.6
23	—10.5	11.77	0.0850	—44.7	535.6	580.2	530.2
24	— 8.8	11.30	0.0885	—42.9	536.1	579.0	528.8
25	— 7.2	10.88	0.0919	—41.3	536.5	577.8	527.4
26	— 5.7	10.50	0.0953	—39.7	536.9	576.6	526.1
27	— 4.2	10.13	0.0987	—38.1	537.4	575.4	524.9
28	— 2.7	9.78	0.1022	—36.5	537.8	574.3	523.7
29	— 1.3	9.47	0.1056	—35.0	538.2	573.2	522.5
30	+ 0.1	9.17	0.1090	—33.6	538.5	572.1	521.3
31	1.4	8.90	0.1124	—32.2	538.9	571.1	520.2
32	2.7	8.64	0.1158	—30.8	539.3	570.1	519.1
33	4.0	8.39	0.1192	—29.5	539.6	569.1	518.0
34	5.3	8.15	0.1226	—28.2	540.0	568.1	516.9
35	6.5	7.93	0.1260	—26.9	540.3	567.1	515.8
36	7.7	7.73	0.1294	—25.6	540.6	566.2	514.8
37	8.9	7.52	0.1328	—24.4	540.9	565.3	513.8
38	10.0	7.34	0.1362	—23.2	541.2	564.4	512.8
39	11.1	7.16	0.1396	—22.0	541.5	563.5	511.9
40	12.2	6.99	0.1430	—20.8	541.8	562.6	511.0
180	89.4	1.666	0.6000	64.6	558.8	494.1	439.5
181	89.7	1.656	0.6034	65.0	558.8	493.8	439.2
182	90.1	1.647	0.6068	65.4	558.9	493.4	438.8
183	90.4	1.639	0.6102	65.8	558.9	493.1	438.5
184	90.7	1.630	0.6135	66.2	559.0	492.8	438.2
185	91.1	1.621	0.6168	66.6	559.1	492.4	437.8
186	91.4	1.613	0.6200	67.0	559.1	492.1	437.5
187	91.7	1.605	0.6223	67.4	559.2	491.8	437.2
188	92.1	1.596	0.6266	67.8	559.3	491.5	436.8
189	92.4	1.588	0.6298	68.2	559.3	491.2	436.5
190	92.7	1.580	0.6330	68.6	559.4	490.9	436.2
191	93.0	1.572	0.6362	68.9	559.4	490.5	435.9
192	93.4	1.563	0.6395	69.3	559.5	490.1	435.5
193	93.7	1.555	0.6428	69.7	559.6	489.8	435.2
194	94.0	1.548	0.6460	70.1	559.6	489.5	434.9
195	94.3	1.541	0.649	70.5	559.7	489.2	434.5
196	94.6	1.533	0.652	70.8	559.7	488.9	434.2
197	94.9	1.526	0.655	71.2	559.8	488.6	433.9
198	95.2	1.519	0.658	71.6	559.8	488.3	433.6
199	95.5	1.512	0.661	71.9	559.8	488.0	433.3
200	95.9	1.504	0.665	72.3	560.0	487.6	433.0
202	96.5	1.489	0.672	73.1	560.1	487.0	432.3
204	97.1	1.474	0.679	73.8	560.2	486.4	431.7
206	97.7	1.460	0.685	74.6	560.3	485.8	431.7
208	98.3	1.447	0.691	75.3	560.4	485.1	430.5
210	98.9	1.433	0.698	76.0	560.5	484.5	429.8
212	99.5	1.419	0.705	76.7	560.6	483.9	429.2
214	100.1	1.406	0.711	77.4	560.7	483.3	428.6
216	100.7	1.394	0.717	78.1	560.8	482.7	428.0
218	101.2	1.382	0.724	78.8	560.9	482.1	427.4
220	101.8	1.370	0.730	79.5	561.9	481.5	426.8

TABLE 3. AMMONIA REQUIRED IN EXCESS OF THAT NEEDED FOR THE EXPANSION COILS

Capacity of Machine	Pounds of Ammonia	Capacity of Machine	Pounds of Ammonia
10	200	100	580
15	250	150	680
25	350	200	780
35	400	300	1,080
50	450	400	1,380
65	500		

As 40.6 lb. is needed for the expansion coils and 450 lb. for the compression side of the system, 490.6 lb.—say five 100-lb. drums—is required for the working charge. Of course, during operation or when ammonia is added to the system, the receiver glass will show whether the charge is too little or too much. Also, with too little in the system, and on a heavy load, there will not be enough liquid to prevent gas from going through the expansion valve. When there is much too great a quantity of ammonia in the system, the receiver will show full and the liquid will begin to fill the condenser. If this condition gets worse, the head pressure will rise.

To determine how much ammonia should be circulated per hour through the expansion coils, and of course through the whole system, we must first find out what is the refrigerating effect of a pound of ammonia expanded between the head pressure and the suction pressure. In our problem the pressures are 25 lb. and 170 lb. gage respectively.

The refrigerating effect is found from the formula,

$$h = H - (t_1 - t) \times S$$

where

h = Refrigerating effect of 1 lb. of ammonia;
H = Latent heat at temperature in expansion coil;
t = Temperature in expansion coil;
t_1 = Temperature of liquid before passing expansion valve;
S = Specific heat of liquid.

Referring to Table 2, giving properties of ammonia at some pressures, we find that for our problem H = 562.6 B.t.u., t = 12.2 deg. F., t_1 = 91.1. The specific heat is 1.1.

Substituting, we have

$h = 562.6 - (91.1 - 12.2) \times 1.1 = 475.81$ *B.t.u.*

That is, theoretically every pound of the ammonia will take up 475.81 B.t.u. per hr.

A ton of refrigeration is equivalent to 288,000 B.t.u. So if we multiply this value by the capacity of the compressor in tons and divide the product by the product of the refrigerating effect of a pound of ammonia times 24 hr., we would get the number of pounds of ammonia necessary to circulate per hour to make the machine run at its rated capacity.

Expressing this as a formula, we have

$$W = \frac{288{,}000\,T}{24h}$$

where

W = Pounds of ammonia circulated per hour;
T = Tons of refrigeration per 24 hr.;
h = Refrigerating effect of 1 lb. of ammonia.

Substituting the values, we have

$$W = \frac{288{,}000 \times 50}{24 \times 475.81} = 1{,}261\ lb.$$

With the foregoing worked out, the applicant should be able to correctly answer the examiner.

Steam Separators with receivers of liberal proportions should be used near engines, to provide a reservoir of steam near-by and to minimize pulsations in the lines.

JUST FOR FUN

Photographing Under Difficulties

A *Power* editor was the first to carry a camera into the power house of a company engaged in the manufacture of war materials. About 300 uniformed guards were employed, and a camera acted upon them much as a red flag affects a bull. Obtaining the necessary pass, the editor was stopped at the entrance of the power house by a guard who demanded a look at the "scrap of paper." An inside patrol likewise wanted a peep and, upon seeing the camera, sent in a call, assisted by a brother guard, for the corporal. Upon his arrival the camera battery suspended action, and the editor, surrendering to overwhelming force, was marched to the detention camp a

prisoner. Fortunately telephone communication had not been cut off during the engagement, and soon orders were received to allow the enemy to return to the trenches and continue the camera bombardment.

The interior pictures were finally taken, and the editor wanted a last one of the cooling pond. With the fever of battle coursing through his veins, he left the protection of the power house only to encounter an outside guard. The overworked pass was produced for the "steen hundredth" time and apparently satisfied the picket. The camera was trained on the spray-pond marine fleet when reinforcements appeared in force, surrounding the invader, who again surrendered. The pass meant nothing to the attacking party, and the camera and gunner were forced to retreat to the protection of the power plant, where through an open window a shot was taken of the cooling pond, while the enemy sent in a second hurry-up call for the corporal of the guard. When he arrived all pictures had been taken, the camera was stored in a bombproof bag, and the editor was making notes. The corporal wanted to take the camera to the guard house, but it was refused him; then it was suggested that the editor go in place of it, but he refused and suggested an exchange of notes or, rather, telephone messages with the imperial office. This was done, and as the offender was an unarmed noncombatant equipped with a proper passport that should have satisfied anyone possessing normal intelligence, he was allowed to finish his work.—Special War Correspondent.

The Production of Coal in Virginia in 1915, according to figures compiled by the United States Geological Survey in coöperation with the Virginia Geological Survey, was 8,122,596 tons, valued at $7,962,934. This is an increase, compared with 1914, of 2 per cent. in quantity, but a decrease of less than 1 per cent. in value.—"Commerce Reports."

Editorials

Foreign Markets and Lack of Shipping Facilities

Attractive markets for power-plant machinery and supplies are opening up in foreign countries, particularly in South America. An advertiser in one of our publications, who formerly did a business of only eight thousand dollars a year in Brazil, is now sending to that country alone goods worth nearly fifty times as much. Manufacturers in the United States are, however, unable to take advantage of this demand on account of the lack of shipping facilities. Instances have recently occurred in our field where large and profitable orders could not be filled because it was impossible to deliver the goods. One firm in New York has orders to the amount of half a million dollars which, for this reason, it cannot fill.

Prior to the war freight rates to Buenos Aires were about four dollars a ton. The average purchase price of a freight steamer is about fifty dollars per ton. Recent freight rates have actually exceeded fifty dollars per ton, so that the freight rate for a single thirty-day trip is now greater than the purchase price of a steamer before the war. This condition is deplorable, as the satisfactory filling of these orders would mean not only present profit, but the establishment of a permanent outlet for our machinery, apparatus and supplies. There are interned in New York and other American ports bottoms which could be profitably employed in this service, and some method ought to be found whereby they could be put into that service to a general all-around advantage.

Putting the Central Station in a Bad Light

Commenting upon the proposal by one of the metropolitan dailies that flood illumination be applied to the Statue of Liberty, Arthur Williams, of the New York Edison Company, speaking at the National Electric Light Association Convention, protested against indorsement of any plan that might include isolated-plant service. He was willing that the association approve the project only on condition that a submarine cable be laid from the mainland and service be supplied by one of the neighboring public-utility corporations.

There is at present an independent plant operated in connection with the statue and the Government station on Bedloe Island. Whether the power for the proposed illumination could be supplied better by an addition to this plant, to be contributed by popular subscription as has been suggested, or by the laying of a submarine cable, is a matter to be decided purely on its merits. Offhand, it would seem that the independent plant would offer the more economical solution; extenuating circumstances, however, may favor the cable. In any event it is an engineering problem and should be handled as such.

The National Electric Light Association, representing as it does the central-station interests of the country, is concerned mainly with benefiting its company members both technically and commercially. It can hardly afford, however, to countenance poor engineering in return for slight commercial gain. As was pointed out, illumination of the Statue of Liberty would likely be followed by the illumination of numerous monuments in other cities, which would redound to the benefit of legitimate central-station business throughout the country. What does it matter then if one statue, because of local conditions, is supplied by a private plant, if it is to be the means of bringing more business to the industry as a whole?

Any such narrow policy as advocated by Mr. Williams would, to use the expression of one speaker, be "putting a fly in the butter."

Service First--Responsibility Second

Nothing marks the engineer of sound judgment better than efficient work in restoring normal service when there has been a breakdown or interruption. Perhaps nothing more clearly indicates a man's present unfitness to lead than a disposition to fix blame upon others before all possible steps are taken to resume full operation. When an interruption comes, be it small or great in terms of equipment or service volume affected, the question of fixing responsibility is one of the first things that naturally rushes into the mind. Some idea of how the accident or other emergency happened may be immensely important in restoring service at the earlist possible moment, particularly where equipment only partly goes wrong; but to know what to do to clear the situation inside the station or on the outside lines is very different from the ability to conduct a sort of homespun court-martial inquiry looking toward the elucidation of all the facts and inferences.

It would be foolish to decry the importance of finding out everything possible about an operating slip-up, in its proper season, but the amount of money tied up in modern plants and their related service is too great today to justify investigation of personal responsibility before action, speaking in general. The experienced chief knows well that at the moment an accident or interruption occurs, the first duty, once life and property are safeguarded to the utmost, is to get things going again. He appreciates that often his subordinates are too much occupied in helping secure this result to be interfered with by questioning about the acts of individuals. Even the man who may be the cause of the difficulty may rise to new competence in the face of the emergency and, by the display of hitherto unsuspected qualities, go far toward retrieving his mistake by his ability in putting the plant back on the line once more. In due time, of course, he must expect to make full explanation to his superior; but the point to be emphasized here is that when a man has made an error and is doing all that is possible to restore the service, there is no time in which to assail his lack of skill or judgment and thereby perhaps "rattle"

him all the more. The chief engineer who forgets everything else in trying to find out who is to blame is not to be compared with the man who takes in the situation almost at a glance, cuts his inquiries to the absolute minimum necessary and takes the immediate steps required without a word of censure, leaving that for the later session with his subordinates or associates, when things have calmed down enough to yield a fruitful inquiry.

To omit every effort to ascertain the cause of a disturbance and to bring the matter home to the man responsible, if such there be, is to run great risk. Such an inquiry should be conducted just as soon as the plant has been put back in normal condition, or at least when all that can be done to that end has been accomplished. Otherwise the men more closely in touch with the occurrence, which often takes place when the chief is out of immediate contact with the situation, may lose some of their recollection of what happened. It is often difficult to recall enough details associated with a station upheaval to satisfy the needs of an engineering investigation of causes, and where the causes of disturbances are more or less obscure, it is of course all the harder.

Too often the station log sheet lacks essential information of this kind, the operator contenting himself with the mere statement that a certain interruption occurred and lasted for so many minutes or seconds. But making notes about an emergency should never be allowed to impede progress toward restoring normal conditions, and if every man on the payroll strives to train himself to recall as much detail as possible about unusual occurrences which lead to interruptions or to other serious trouble, comparatively little will be lost when the inquiry by the chief, and perhaps by his superiors, is begun to fix responsibility and take such steps as may be necessary to avoid the repetition of the disturbance.

The National Electrical Safety Code

The recent conference on the National Electrical Safety Code, between the Bureau of Standards and representatives of the American Institute of Electrical Engineers, the National Electric Light Association and a number of large operating companies, appears to have resulted in a deadlock on some important points.

The code has now been in preparation over two years; several tentative forms have been printed and about thirty conferences held. It is contended by some that the Bureau of Standards has taken the initiative without authority; and while it can do no more than recommend, the public-service commissions will undoubtedly adopt the provisions of the code and thus work hardships on some of the operating companies. However true this assertion, it is undeniable that the fundamental aim is most commendable. That there is need of an enforceable electrical safety code was manifest in the recent report of the Accident Prevention Committee of the National Electric Light Association, which pointed out that, although creditable work is being done by some of the larger companies, the majority of the member companies show lack of interest in accident-prevention work.

If such work is to be carried through successfully, it is desirable that it be undertaken by some responsible and disinterested party, and the Bureau of Standards, as a central body, is better qualified to do this than individual public-utility commissions in the various states.

One of the principal differences of opinion between the operating companies and the bureau relates to prescribed construction on transmission lines and crossings. The companies claim, with some justification, that the requirements would make the costs prohibitive, especially as applied to extensions into new territory; that it would be practically impossible to follow exactly the provisions of the code, and in view of this any accidents that might result would react heavily upon them.

In any undertaking of this kind it is impossible to please all and each side must compromise. The operating companies have their peculiar problems and must often view new work and restrictions from several angles, some of which may not be apparent to the representatives of the bureau. The latter are aiming to make electrical construction one hundred per cent. safe, but may have to be satisfied for the present with ninety per cent. The suggestion that the code be tried out for a year offers a solution that would permit more intelligent revision at the end of that period.

Meeting Exigencies of the War

Out of the devastation and calamities of the European War there may come, aside from military and political considerations, some compensation in the form of greater industrial organization and higher development along certain lines of engineering. The Diesel engine and the aëroplane, for instance, have possibly advanced more during the past eighteen months than they would have in ten years of peaceful progress.

Germany in particular, as a result of the enforced economic situation, will probably emerge from the war much better equipped to cope with competition and the pending abnormal industrial conditions. The article on page 831 shows briefly how some of the exigencies have been met, especially as regards the substitution of other metals for copper in electrical machines, switchboard equipment, etc. Iron, zinc and certain alloys appear to have taken the place of copper for such purposes, although their conductivities are considerably less. This feature has heretofore debarred them from competition with copper, but it is possible that as a result of the lessons learned they may in the future permanently replace copper for certain parts and thus have an important bearing on the design of electrical apparatus.

Like the Diesel engine in the submarine, the whole story will probably never be known, save to a few, but the effect will be stamped upon the future.

Like the poor, the man who closes a three-inch valve with a twenty-four-inch wrench will always be with us.

Metallurgists say nitrogen in iron and steel has a bad influence. It isn't worth much in flue gas either.

Feather valves are now being applied to ammonia compressors, but even these will not allow of such high speeds that the machines may be said to fly.

If you do not already, you will eventually appreciate that marine engineers make some of the best found in stationary practice.

Correspondence

Home-Made Pump-Speed Regulator

Referring to Mr. Brewer's "Home-Made Pump-Speed Regulator," in the issue of May 2, page 631, this idea is used in plants with individual boilers when not equipped with feed-water regulators. It should prove satisfactory where the water tender or fireman has the pump directly at hand and can regulate its speed to suit the feed.

The statement in regard to an $80 regulator seems rather off, unless a lot more was paid for the regulator than was necessary. Two and no doubt three first-class pump governors could be purchased for $80. The throttle-valve seat and disk should not cut with a governor in service, for it should be opened wide and the governor allowed full control. JOSEPH KROS.

Springfield, Ill.

The Gage Showed False Back Pressure

The engineer in a small refrigerating plant was having considerable trouble with air in the system. He was unable to account for the presence of the air and had to purge the condenser once a week.

Some of his engineer friends and various ammonia salesmen laid it to the ammonia, which they claimed was of inferior quality and was deteriorating into foreign gas. Acting on their advice, he pumped the whole charge out during the winter season and sent it to an ammonia manufacturer. The expansion coils, the condenser, the whole system in fact, was cleaned thoroughly. All the joints were gone over, tightened and tested with 200 lb. air pressure, and all proved to be tight. It was then tested under 25 in. vacuum and also shown to be tight, as it maintained this vacuum for several hours after the pump was shut down.

When the plant was started in the spring, he still had as much trouble as ever. The rods had been turned up during the winter, new packing had been put in, and there was not a smell of ammonia leaking out, so there could be no possibility of air being drawn in at the glands. Besides, he carried 5 lb. back pressure and pumped down very seldom, and then only to zero, so there was no possibility of air being drawn in anywhere.

The engineer at this time secured a better position, and a new man was put on the job. The first day the latter got a surprise. The machine was equipped with a small hand oil pump for pumping oil to the lanterns of the stuffing-boxes. The new engineer filled the pump in the morning and pumped a little oil into the stuffing-boxes. A couple of hours later, when he came back to pump in more, he was astonished to find the pump empty. He filled it again, pumped oil to the stuffing-boxes and in another hour found the oil gone again. He then filled the pump and watched it. The oil slowly disappeared. The valves in the lines leading to the stuffing-boxes were closed. Besides, there was 5 lb. suction pressure, so it was impossible for the oil to be drawn in from this cause. When the oil was gone, there was a slight gurgle and then a gentle hissing sound in the pump. The engineer was convinced by this time that air was being drawn in through the pump. He first pumped out the machine, shut down and removed the bonnet from the stop valve in the oil line. A large piece was broken out of the disk of this valve. Even so, there should not have been any air drawn into the machine because the suction pressure was above atmosphere. He finally came to the conclusion that there was something wrong with the suction-pressure gage. When he had it disconnected, the hand, instead of going back to the zero point, went up to 8 lb. Instead of 5 lb. back pressure, he had been running with about 6 in. vacuum. The vacuum had drawn the oil and air in through the broken valve. The reason the old engineer had failed to notice this was that he had been in the habit of pumping the whole pumpful of oil into the stuffing-boxes every time he put any in. Consequently the pump was always empty. After a new valve had been put in, the suction-gage indicator set back to zero, and the suction pressure brought up to where it should be there was no further trouble. THOMAS G. THURSTON.

Chicago, Ill.

Galvanized Pipe for Ammonia

D. L. Fagnan's letter on "Galvanized Pipe for Ammonia," pages 666-7 of the May 9 issue, strikes me as worthy of discussion. I have had occasion to observe the effect produced when galvanized pipe is used for ammonia condensers. When taking down some old ammonia condensers, I found that where the hot gas entered the galvanizing had disappeared from the inside of the pipes, but on the pipes subjected to moderate and cold temperatures it was practically intact. In the hot pipes the galvanizing had deposited in dust form in the bottom of the pipes.

In another plant I observed practically the same effect, but did not notice that galvanizing aided the deterioration of the ammonia. From my own experience I know that deterioration is caused, or at least aided, mainly through air, low grades of oil used for compressor lubrication and water getting into the system. On one occasion salt brine had got into the system, and there was no end of deterioration for about six months.

The bad effects said to result from mixing ammonia from different manufacturers is, in my opinion, much over-estimated. Noncondensable gases will cause the ammonia vapor and liquid to be alternately trapped and released. This is what produces the erratic working of the compressor valves and makes the piston rod get hot and cold.

The trouble with these gases is that one cannot with any certainty locate their hiding place. They are not always in the top of the condenser as supposed. If this gas were air only, it would not be in the top of the con-

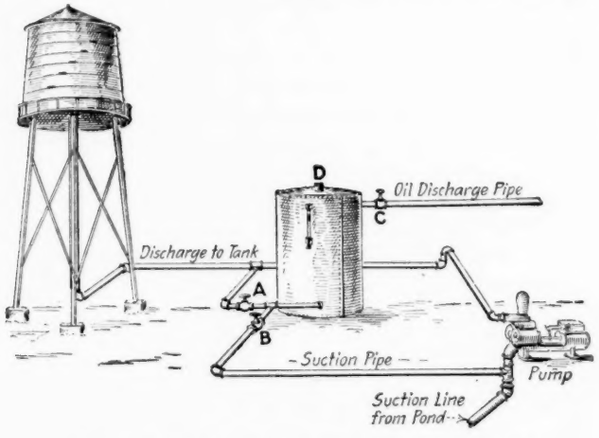
denser, because air is heavier than ammonia gas. The gases play the same tricks as air or gas in a heating system, but the latter has the advantage of vents or purge valves in convenient places, which is not possible in the ammonia system. One of the best ways to avoid trouble is to keep the system properly charged with ammonia at all times.

WILLIAM SCHLIEMANN.

New York City.

Clean, Handy Oil Storage

At the plant of which I am engineer there was formerly a great waste of oil and much trouble with barrels. We first considered installing a low-level tank and using air to force the oil to the engine room, but found it difficult to keep the air pressure up all the time, so decided to use the water-tank pressure instead, and it worked out very



NECESSARY TANK AND PUMP CONNECTIONS

nicely. The oil tank is of 750 gal. capacity and is below ground. The oil is brought to the plant in a tank wagon and put in the tank at *D* (which is afterward plugged). Valves *A* and *B* are closed while the tank is being filled, after which *A* is opened. The oil is then under pressure and is discharged through the line to the engine room when valve *C* is opened.

When water comes in sight in the gage-glass, valve *A* is closed, shutting off the water, valve *B* is opened and the water is then drawn out by the pump, a stick being used to tell when there is enough drawn out. Valve *B* is then closed, and more oil is put in as before.

Coffeyville, Kan. H. P. DOERING.

Comparative Quality of Oil

The consumer always wishes to purchase the best oil for the least money, and the quality of a certain grade of oil being undeterminable to the eye, resort is had to practical, physical and chemical tests, such as specific gravity, degree Baumé, weight per gallon, refractive index, viscosity at 70, 120 and 180 deg. F., flashpoint, burning point, congealing point, color density, carbon content and free acid. There are manufactured in all commercial products such as oils of a supposedly certain grade, superior and inferior grades, sometimes perhaps beyond the control of the maker. Usually, however, the maker knows to whom he had best ship the inferior grade; his knowledge of this being based upon the fact that there is no inspection at the point where the inferior grade is shipped. I recall an instance where the price asked per gallon of 90-per cent. quality oil was the same as that paid for supposedly the same grade which had 64-per cent. quality.

After having carried on extensive tests of oils for gas and gasoline engines that were submitted by about twenty companies, I found it desirable to establish a rating or, percentage of quality. This was arrived at as follows:

RULE FOR RATING OILS

A. Viscosity at 120 deg. F. × 1
Viscosity at 180 deg. F. × 10
Flashpoint × 1
Multiply and add the products.

B. Congealing point × 1
Free acid × 100
Carbon content × 100
Multiply and add the products.

Subtract B from A, divide by 5 and then multiply by the specific gravity. The result is the percentage of quality.

If the viscosity at 70 deg. F. is considered, divide 100 by this figure and add to the quantities in A.

Example:

Viscosity at 120 deg. F.	1 ×	2.09	= 2.09
Viscosity at 180 deg. F.	10 ×	1.22	= 12.22
Flashpoint	1 ×	402	= 402.00
			416.31
Congealing point	1 ×	20	= 20.00
No free acid			
No carbon content			
			A — B = 396.31

396.31 ÷ 5 = 79.25
79.25 × 0.901 = 71.40 per cent.

Out of ten heavy crank-case oils computed in this way the quality varied from 96.94 to 50.94; of twelve crank-case and gas-engine oils the quality ranged from 80.5 to 56.94; of nine motor and automobile oils, from 73.27 to 60.24; and of three common paraffin oils, from 74.30 to 44.97.

It is of course necessary to change the factors in the rule to suit the characteristics of various oils, such as cooling, thread-cutting, spindle, dynamo and special oils; or by using a fixed rule, one will see that oils with various characteristics are divided into classes after analysis.

Mansfield, Ohio. G. A. SMITH.

Flickering Lights

Replying to F. C. Bulsara's inquiry in the Apr. 18 issue regarding flickering lights, will say that I experienced similar trouble, which was first noticed when metal-filament lamps were introduced some years ago. I began to investigate and found the cause at the commutator and brushes. The flickering was not apparent when carbon lamps were used, although the cause existed just the same.

I first called in the makers of the machine, who attributed the trouble to the engine governor. I soon showed them their error, for by shifting the brushes slightly forward or backward flickering would cease at that particular load. When the load fell off, I shifted the brushes back to a point where there would be no flickering; this proved that the fault was not in the governor. I then set to work and sandpapered the brushes and commutator, and the flickering ceased.

Millvale, Penn. H. G. BAER.

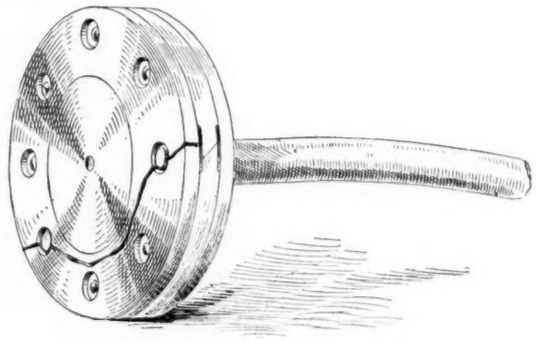
I read with interest F. C. Bulsara's letter in the Apr. 18 issue and would say that I have experienced similar trouble in different forms. He did not mention whether his machines were direct-connected or belt-driven and whether he had low- or high-speed engines.

I have had trouble with flickering lights because of the load being unbalanced, which overloaded the balancing set and caused it to spark badly; also from the shaft governors sticking at the pivot and the gland being set up too tight on the valve-rod packing. If Mr. Bulsara has belts, he should look to them for hard spots, which will cause the belt to slip slightly. I take it for granted that his generator commutators are in good shape.

Danvers, Minn. FRANCIS J. DOYLE.

Broken Follower Plate

A peculiar accident happened on Friday, Mar. 24, at about 7 p.m., when a 2,000-hp. cross-compound condensing horizontal Corliss engine suddenly developed a serious thump in the high-pressure cylinder, which continued for



BROKEN ENGINE PISTON

about 15 min. On the following Sunday morning the cylinder head was removed and it was found that the follower plate was broken as shown in the illustration. All of the follower-plate bolts were broken off except two where the break occurred. Marks on the cylinder head showed where a bolt head had been caught between it and the piston, which probably broke the follower plate. The piston rod was also bent. C. A. DUVALL.

Monessen, Penn.

Economy of the Univalve Unaflow Engine

I notice on page 727 of the May 23, 1916, issue a reproduction of some illustrations taken from the catalog of the "Univalve Unaflow" engine, built by the Skinner Engine Co., of Erie, Penn., and I regret that the builders of this engine did not know that this description was to be published, or they would have furnished some interesting data as to what the engine has been accomplishing in actual practice.

This engine is built only in small sizes, 160 hp. being the largest, and the design is the result of the application of the steam-tight-valve feature to the small-clearance noncondensing unaflow principle, which have been developed by the company in larger engines, known as the "Universal Unaflow."

The description given is in error regarding the auxiliary exhaust ports being controlled by separate valves. They are controlled by the admission valve in conjunction with the piston. One valve controls the two admission and the two auxiliary exhaust ports, hence the reason for naming it the "Univalve Unaflow." The piston itself, of course, controls the commencement of compression by covering the auxiliary exhaust ports. These ports are necessary so that the engine may operate under noncondensing conditions with small clearances.

The object of the builders of the "Univalve Unaflow" engine was to produce a more economical small steam engine than power users have heretofore been accustomed to, in order to stop the encroachments of the gas engine and the electric motor driven by central-station current.

To demonstrate that small plants equipped with "Univalve Unaflow" engines could be operated successfully in comparison with central-station current rates, the builders made a contract to install a 72-kv.-a., or approximately 50-kw. engine, generator, exciter, switchboard, etc., and agreed to take their pay for this equipment in the saving it effected against the central-station current rates in two years and five months, plus an initial cash payment of $1,000.

This plant is effecting a monthly saving of $150 against the central-station rates, on a consumption of 8,320 kw.-hr. The engineer's wages of course are not taken into consideration, as the purchasers were forced to employ an engineer whether they purchased their current or made it. There was no charge, however, made for the use of exhaust steam.

On account of the fact that the "Univalve Unaflow" engine has a flat steam-consumption curve—that is, its economy at one-fourth load in pounds of steam per i.hp.-hr. is approximately as good as at full load—it is a very desirable engine for plants that operate on light loads during part of the day.

One of these engines replaced a well-known make of pressure-plate-valve counterflow engine, being direct-connected to the same generator, and without any other change in the plant a 24-hr. test run showed that the "Univalve Unaflow" engine had effected a saving of 52 per cent. in fuel with an increase of 34 per cent. in the load.

In this particular plant there was an 18-hr. light-load period which of course was very favorable to the "Univalve Unaflow" engine on account of its flat steam-consumption curve. A. D. SKINNER.

Erie, Penn.

Oil from Cleaner Blisters Tubes

In a water-tube boiler plant trouble was experienced with blistered tubes. It was at length observed that the tubes showed signs of distortion soon after cleaning, which made the case doubly puzzling. The belief was that sediment was responsible.

A solution of the near-mystery was accidentally found one day when the engineer chanced to see the tube-cleaner operator oiling the rotor of the cleaner for the day's run. The latter explained that he always used plenty of cylinder oil, as a little overflow down the inside of the tube made the machine pass through the tube easier.

The rotor being entered at the top end of the tube (a Stirling boiler) the oil had followed along to the bottom, flowing, in some boilers, into the bottom drum. It seems hardly possible that a man would do such a thing; yet here it was, and the evidence was all too plain. How many other turbine operators flood the rotor with oil?

Philadelphia, Penn. EDWARD T. BINNS.

Fuels for the Oil Engines

When Diesel began his work, he was not content with the idea of using liquid fuel, but proposed to use pulverized coal as well. This proved inadvisable because of the ash, which must have a bad effect upon the cylinder even if a valve could be made that would handle the coal. There seems to be no good reason, however, why all liquid fuels cannot be used in the oil engine.

One thing for consideration is the ignition points of the different fuels. There is good reason to believe that this point is practically constant for all fuels. If this be true, the only limitation to the fuel for use in the oil engine is that placed by high viscosity, and this seems to be overcome when the fuel is heated and fed through a valve that does not require a water jacket.

The first reason for believing that all fuels have practically a constant point of ignition is in the fact that the chemical composition of all hydrocarbons is practically constant. The carbon content varies from 80 to 90 per cent. and the hydrogen content from 8 to 15 per cent. Even the solid residues from fuel of an asphaltum base come within this range. The only reason for questioning this statement of ignition point would lie in the fact that these fuels have the carbon and hydrogen united in different series of hydrocarbons.

By ignition point is meant the point which, if the fuel be brought in contact with air of the same temperature, ignition will take place spontaneously. This should not be confused with the flashpoint, which is simply the point at which a vapor is given off that readily forms an explosive mixture.

The best practical proof of this common ignition point lies in the fact that the oil engine has shown that heavy fuels will ignite at around 1,000 deg. F., and the condition at which a vapor engine will continue to run even if the ignition means be discontinued, is clearly over 800 deg. This can be determined by a calculation of the amount and temperature of the products of combustion remaining at the end of the discharge stroke and by an approximation of the temperature of the charge when it has been drawn into the cylinder over the hot head.

If the ignition points of all hydrocarbons be practically constant, the problem narrows to one of physical properties. The fuel must be reduced to a liquid condition by the addition of heat, and the fuel valve must be so designed that it will not depend upon the size of the passage for producing a perfect spray; for if it does, a valve that will use a heavy fuel with good results will deliver a lighter fuel into the cylinder in the form of a slug of oil. On the other hand, a differently proportioned valve will use the lighter fuel with good results and fail on the heavy fuel. Consequently the use of all grades of fuels in the same engine simply depends upon the design of a fuel valve in which the fuel is positively presented to a current of fast-moving air. It should be easily possible to design such a universal fuel valve, the point being, is it worth while to produce such a fuel-feeding arrangement?

In the description of a marine oil engine of the Werkspoor type, the statement was made that heavy fuel can be used, but the difference in cost between this oil and a lighter oil is so slight that there is but little use in taking the trouble with the heavier fuel. This is a confession of unreliability with the heavy oil. In engines of large power, which are coming to the front rapidly, 0.10 of a cent a gallon will amply repay the trouble of designing a positive and reliable means for using the lowest grades of fuel. Those engines that can use the poorer grades of fuel with absolute reliability will drive out the engines using the lighter and more expensive fuels. In fact the chief advantage of the heavy-oil engine lies in the hope of using all grades of liquid fuels. Purchasers of oil engines should look into this point very carefully before buying. It may be that owing to transportation costs the better oil will be as cheap as the poorer oils in some localities, but this is a condition that will be overcome to a degree at least as soon as the engine is produced which shows a broad market for the lower grades of oil.

Quincy, Mass. JOHN F. WENTWORTH.

Bricking-in Tubular Boilers

The methods of inclosing the sides of the shells of tubular boilers, of making them gas-tight where the brickwork meets the shells, have not been as widely discussed as some engineers wish. The most familiar way is well known; but it is the new methods that one would like to see illustrated, described and commented upon.

New Orleans, La. ORMAND WILSON.

Another Safety Sign in Use

Occasionally safety-first suggestions and signs appear in *Power,* so I am sending a photograph of one in use in our plant. This one is painted on the end wall of the engine room, where it can be seen from any part of the room, and probably it has more good effect than is generally supposed.

LARGE SIGN PAINTED ON WALL

WILLIAM T. O'REGAN.

Haverhill, Mass.

Cost of Power-Plant Scales

The following table gives the cost of installing a set of 10-ton scales of the platform type in a Massachusetts power plant. The scales are mounted on foundations of stone and brick.

Excavation	68 cu.yd.	@	$0.75	$51.00
Stone	8.2 cu.yd.	@	6.00	49.20
Brick	8.2 cu.yd.	@	23.00	188.60
Concrete (4-in. floor)	168 sq.ft.	@	.18	30.24
Steel	132 lb.	@	.06	7.92
Cast iron	40 lb.	@	.04	1.60
Timber framing	2.2 M. ft. b.m.	@	45.00	99.00
One ten-ton Howe scale				200.00
Total				$627.56

Cambridge, Mass. H. S. KNOWLTON.

Inquiries of General Interest

To Convert Parts per 100,000 to Grains per Gallon—When impurities in feed waters are quoted as parts per 100,000, how are the proportions reduced to grains per gallon? J. R.

One United States gallon of water at 62 deg. F. weighs 8.3356 lb., or 8.3356 × 7,000 = 58,349.2 grains; hence to convert parts per 100,000 to grains per gallon, multiply the number of parts by 58,349 ÷ 100,000 = 0.58349.

Stroke and Throw of Engine—Which is the forward stroke of an engine, and when does an engine throw over or throw under? C. H.

As the piston leaves the head end of the cylinder and approaches the crank, it is said to be making its forward stroke, and the reverse motion is called the return stroke. An engine is said to "throw over" when the crank is above the center line during the forward stroke and to "throw under" when the crank is below the center line during the forward stroke.

Equivalent Evaporation from and at 212 Deg. F.—An actual evaporation of 8.4 lb. of water was obtained per pound of coal when generating steam at 120 lb. gage pressure from feed water at 190 deg. F. What was the factor of evaporation and the equivalent evaporation from and at 212 deg. F.? J. J. N.

By reference to the steam tables it may be seen that a pound of dry saturated steam at the pressure of 120 lb. gage, or 135 lb. per sq.in. absolute, contains 1,191.6 B.t.u. above 32 deg. F. As each pound of the feed water contained 190 — 32 = 158 B.t.u. above 32 deg. F., for its conversion into steam each pound of the feed water received 1,191.6 — 158 = 1,033.6 B.t.u. The heat required for evaporation of one pound of water from and at 212 deg. F. is 970.4 B.t.u.; hence, under the conditions, the factor of evaporation was 1,033.6 ÷ 970.4 = 1.065, and the evaporation was equivalent to 8.4 × 1.065 = 8.946 lb. of water evaporated from and at 212 deg. F. per lb. of coal.

Relative Operation of Pump at Higher Altitude—Will a pump with 4-ft. suction-lift work as well at a place whose elevation is 7,500 ft. as at sea level? W. J. B.

At an altitude of 7,500 ft. above sea level the atmosphere exerts a barometric pressure of about 22.5 in. of mercury, or about 11 lb. per sq.in., while at sea level the barometric pressure is about 30 in., or 14.7 lb. per sq.in. Hence at the greater elevation there would be less suction pressure and for the same size of suction pipe and passages there would be less suction flow available. When operating within the lower rate of suction delivery and discharging at, above or below atmospheric pressure, the power required would be the same for the same total lift, rate of discharge and other conditions.

Arrangement of Superheaters—How are superheaters arranged for superheating steam generated in power boilers? W. L.

The purpose of a superheater is to raise the temperature of the steam higher than that at which it is formed. For that purpose the steam to be superheated must be removed from the presence of water. Hence superheaters are so arranged and located that the steam will have to pass through them and become heated on its way from the boiler to the engine. Rise in pressure is prevented by the engine taking a continuous supply. Superheaters usually consist of coils of pipe placed in the path of the hot flue gases on their way from the boiler to the chimney or of coils placed inside the boiler setting so they may receive heat from the furnace gases before, after or during the time the heated gases are swept over the heating surfaces of the boiler; or the superheater is placed in a setting independent of the boiler and separately fired.

Air Required for Combustion—What weight or volume of air should be supplied per minute to a boiler furnace having a grate 6x7 ft. on which 22 lb. of coal is burned per hour per square foot of grate? G. W. M.

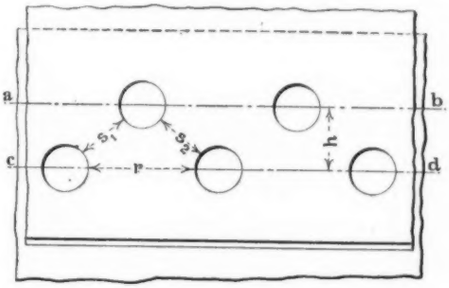
Most fuels for their combustion require between 11 and 12 lb. of air per pound of the fuel. It is necessary, however, to admit more air to the furnace than actually required, to make sure that each atom of carbon will be brought into contact with an abundance of oxygen. The excess air thus introduced is sometimes called air of dilution. The amount of excess air necessary for best results depends on the draft, a weaker draft requiring more air of dilution. For obtaining best results with natural draft it is customary to supply about twice the amount of air actually required for combustion and with forced draft about 1½ times as much. Allowing 24 lb. of air per pound of fuel and combustion at the rate of 22 lb. of coal per hour per square foot of grate, with a grate 6x7 ft. there would be $\frac{(6 \times 7) \times 22 \times 24}{60} = 369.6$ lb. of air required per min. Assuming the air to be admitted at a temperature of 62 deg. F., each pound of air would occupy a space of 13.14 cu.ft. and under the conditions the volumes of air supplied would be 369.6 × 13.14 = 4,856.5 cu.ft. per min.

Quality of Steam Shown by Separating Calorimeter—In using a separating calorimeter the glass gage showed that 0.17 lb. of water was collected during the time the weight of condensing water had increased 2 lb. 5 oz. What was the quality of the steam sampled? R. C.

As the weight of dry steam condensed was $2\frac{5}{16}$, or 2.3125, lb., the total weight of steam sampled was 2.3125 + 0.17 = 2.4825 lb. and the quality or dryness was 2.3125 ÷ 2.4825 = 0.93, or 93 per cent.

Relative Strength of Ligaments—In a staggered-riveted joint, how is it determined whether there is sufficient distance between the parallel rows of rivets to prevent tearing of the diagonal ligaments before tearing takes place between the rivets of the same row? W. C.

It has been well established by experiment that if the rows are wide enough apart to afford 30 per cent. more net section of plate material along any broken diagonal through the rivet holes than from rivet to rivet of the parallel rows,



rupture will be confined to material between the rivets of the same row. Where, for instance, ab and cd, as in the figure, are center lines of parallel rows of rivets, rupture will be along the line r if the spacing h is sufficient to obtain $s_1 + s_2 = 1.3$ r. For ordinary staggered riveting in which $s_1 = s_2$, the usual practice is to provide a sufficient distance, h, between the rows to obtain for s_1 or s_2 a value greater than $\frac{2}{3}$ r.

Greater Loss of Water Level with Steam Shut Off—We have a Scotch marine-type boiler for supplying steam to laundry machinery. The condensate is gathered in a closed receiver from which it is returned to the boiler by a tilting return trap. We bank the fire at night, leaving 50 to 60 lb. of steam on the boiler. Other conditions being the same, what would cause the boiler to lose more water over night when the steam is cut off the plant at the boiler than when it is left turned on? J. F. L.

With steam shut off the trap is inoperative and, probably from leakage of the trap's check valves, the boiler water backs up through the trap to the receiver and its connections. Under such conditions, unless there is some place for drainage, the boiler water might fill the system all the way back to the boiler steam stop valve. The remedy would be to place a stop valve in the trap discharge line to the boiler and keep it closed when the steam supply is shut off from the system.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Ohio Engineers' Meeting at Cleveland

The Station Operating Committee of the Ohio Electric Light Association held its second conference for station-operating men on May 18 at Cleveland in the assembly room of the Cleveland Electric Illuminating Co. Three interesting papers relating to power-plant operation were presented, and each in turn was given an informal discussion by the men on the firing line. The first topic, Electric Versus Steam Auxiliaries, was presented by W. H. Aldrich, of the Cleveland Electric Illuminating Co. Three things must be considered in the requirements for motive power for station auxiliaries: Reliability, economy, and adaptability to location, service and general working conditions. The question of reliability is paramount, and for that reason steam-driven auxiliaries are generally used where interruption of electrical service or serious overload on the main generator (the usual source of supply of current to electrically driven auxiliaries) would shut down or cripple the service of such motor-driven auxiliaries. In case of a complete shutdown, or tripping off the line of all main generators, the steam-driven auxiliaries would not be affected at a moment when the time of the operator is valuable, whereas the electrically driven auxiliaries would stop, and the entire equipment of the station would not be started up without the aid of outside current or an auxiliary steam-driven generator. In the case of heavy overload on the main generators, with the resulting drop in electrical pressure, the steam-driven auxiliaries would be unaffected, while electrically driven auxiliaries would slow down at just the time when they would be called upon to carry extra load. This is particularly true of the condensate pump, the vacuum pump and the circulating-water pump, for the steam consumption per kilowatt-hour of an overloaded turbo-generator is well known to rise rapidly over that of its normal capacity.

The effect of a drop in pressure on alternating-current motor-driven feed pumps would not be so serious, for it is common practice to check down temporarily the feed water in the boiler room, when confronted by a sudden demand for steam; nor would the effect be felt seriously on a motor-driven stoker, for the nature of the service would require a direct-current motor with a wide range in speed, controlled by a rheostat.

SEPARATE CURRENT SUPPLY FOR AUXILIARIES

Where the station is large enough to warrant installing a generator to take care of the auxiliaries, the use of motor drive is comparatively safe and has the added advantage of taking up less space, requiring less piping, and as a general proposition less attention.

As to efficiency, the first cost of electric and steam motive power for auxiliaries is about the same per horsepower. The cost of installing the motor-driven apparatus is much less, with its saving of steam pipe and covering and of room taken up by piping. The small steam turbine is notoriously inefficient and could stand no comparison with the motor for economy of operation were it not for the utilization of the exhaust steam for heating the feed water. Professor Hirshfeld, chief of the research department of the Detroit Edison Co., is authority for the statement that about 18 per cent. of the heat in the coal is converted into useful electrical energy in the most efficient steam plants operating under the best conditions, and the average performance of a large central station is nearer 13 per cent. With the auxiliaries electrically driven from the main unit, about 0.9 of 1 per cent. of the total heat in the coal is taken by them. In the case of steam-driven auxiliaries 12.8 per cent. of the total heat of the coal enters the auxiliaries, 0.83 per cent. is used for auxiliary power, 11.97 per cent. is rejected in the exhaust, and 2.4 per cent. is lost and cannot be absorbed by the feed water, leaving 9.6 per cent. to be returned to the feed water.

In considering the minor advantages and characteristics of the two forms of motive power, it is well to bear in mind the flexibility of steam drive with its capacity for slight adjustments in speed to take care of wear and resultant slip in centrifugal pumps and its greater adjustments in speed to take care of the quantity of water handled by the circulating pump at opposite seasons of the year.

Constant-speed alternating-current motors have been used with considerable success for all auxiliary service except stoker drive, for capacity regulation can be made by throttling in centrifugal pumps and by dampers in blowing apparatus.

In summing up, the practice of using motor-driven auxiliaries entirely in a station has not been generally adopted, chiefly for two reasons: First, the use of exhaust steam from steam-driven auxiliaries in feed-water heaters has been considered to give sufficient economy to warrant the use of steam rather than motor drive; and second, the operation of auxiliary equipment in the station, by current taken from the main generators, introduces a considerable element of uncertainty and unreliability into the service. Those who advocate the use of electrically driven auxiliaries entirely in a power house answer these objections by claiming that the thermal efficiency of the station can be shown to be much higher when motor-driven auxiliaries are used and the heat for boiler-feed water obtained from the flue gases by the use of an economizer. They claim that no arrangement of steam-driven auxiliaries will give just the proper amount of exhaust steam for heating the feed water properly at all loads on the station, and that there will always be periods when there will be not enough or too much steam with consequent waste. There is also a considerable amount of steam piping necessary for supplying the auxiliary engines or turbines, with the incident losses of radiation. The second objection of uncertainty of service they answer by installing in the station an auxiliary steam-driven generator.

DISCUSSION OF DRIVE FOR AUXILIARIES

In the discussion it was recognized that a choice of steam or electrically driven auxiliaries depends upon the conditions in any particular plant and the possibilities of maintaining a heat balance under varying loads. The controlling factor is reliability. It was brought out that a motor is just as reliable and more economical than a turbine or engine drive. The question reverts back to the source of supply. Average practice has shown that variations in voltage on the station bus that would seriously affect the motor or perhaps cause a complete shutdown are more common than fluctuations of like character in the steam supply. On heavy loads even a comparatively small drop in voltage will slow down the motor at a time when it should be giving its maximum power. Steam-driven auxiliaries are usually designed to operate at full capacities on pressures somewhat below that of the steam supply to the main unit, so that they are affected only by large drops in pressure or a discontinuance of the steam supply altogether.

Unless the motor has a source of supply other than the station bus, such as a storage battery floating on the line or an outside connection, in the event of a shutdown there is apt to be serious delay in getting the main units back into service. The ideal condition would be to have duplicate units throughout, one steam and one electrical. The heat balance could then be regulated to a nicety, there would be double insurance against the shutdown of any particular auxiliary and little delay in shifting over from one to the other. The initial cost, of course, would be increased.

The use of economizers and the turning back of auxiliary exhaust into the lower stages of the main turbine units influence the division between steam and electrically driven auxiliaries. A more economical prime mover to drive the auxiliaries, such as a uniflow engine, was suggested, but the consensus of opinion was to go as far with electric drive as the heat balance would permit. Electrically driven auxiliaries might serve the condensers and steam turbines drive the boiler-fed pumps. And even here reserve electrical units might be installed, so that on light loads there need be no waste of exhaust steam.

GAS VERSUS COAL

A. P. Lewis, of the Cleveland Southwestern & Columbus Ry., introduced the topic of "Gas versus Coal" and showed how a change from coal to gas in his plant made a considerable saving. The plant contains 4,000 kw. in cross-compound engine and turbine-driven condensing units. It also has a 500-kw. rotary converter to help out both ways. It has 4,400 hp. in water-tube boilers, an open feed-water heater, and turbine-driven and direct-acting boiler-feed pumps. The peak load runs as high as 5,000 kw. The boilers were hand-fired, while the coal was unloaded on a trestle and wheeled to a point in front of the boilers. On a basis of one man handling 16 tons of coal per day, the cost reduced to 12½c. per ton. With the fireman shoveling 11 tons of coal per day into the furnace, the cost would be 12c. per ton. The cost of removing ashes was figured at 4½c. per ton of coal, so that the total cost for handling coal and ashes amounted to 29c. per ton. The coal used in the plant cost $1.95 per ton, or $2.25 including the handling.

To reduce the cost in the boiler room the installation of coal-handling equipment and stokers was considered. With these additions it was figured that coal costing $1.65 per ton could be burned and the handling, including overhead on the investment, would increase the figure given 21c., making a total cost per ton of $1.86, or a net saving over hand firing of 39c. per ton. This would effect an annual saving of $18,000 over hand firing. Owing to the construction of the building the installation of this new equipment would have been difficult, so that the question of burning gas was considered and a change to this fuel was finally made.

To put it on an equal basis with coal containing about 13,000 B.t.u. per lb., it was figured that natural gas containing 900 B.t.u. per cu.ft. should be purchased at a rate of 10c. per 1,000 cu.ft. For one cent 90,000 B.t.u. in the gas could be purchased and 137,000 B.t.u. in coal costing about $1.90 per ton. To come out on an equal basis the gas then must be about 50 per cent. more economical than the coal.

In the plant under discussion from 25 to 45 cu.ft. of gas has been used per boiler horsepower-hour. They allowed about 60 hp. per burner. Over a period of several months the gas consumption per kilowatt-hour at the switchboard ranged between 33 and 40 cu.ft., or an average of about 37 cu.ft. Over hand firing they saved approximately $1,500 per month, or about the same as the saving estimated for the plant if it had been equipped with stokers and coal-handling equipment. Roughly, they have found 10 cu.ft. of gas equivalent to one pound of coal.

One of the first troubles encountered in burning gas was a severe vibration which shook the entire setting and could be felt in the building. A change in the intensity of draft or slight alterations in the furnace did away with this nuisance. It appears that certain wave lengths in furnaces of certain proportions followed one another in proper succession to build up and amplify the noise to an unusual degree. They have found it more economical to bank the boilers, or in other words, keep them near the steaming point with one or two burners rather than to have to bring up a cold boiler in the morning.

The discussion turned to the impinging action of the flame on the knee of the front row of tubes in a Stirling boiler, the use of checker work, the location of the burners and the advisability of baffling the front row of tubes to provide a large combustion chamber and long flame travel for the gases before coming in contact with the heating surface.

Gas was recognized as an ideal fuel with which boiler efficiencies up in the eighties can be obtained, because of their clean heating surfaces and the ease with which proper air mixture can be secured. A choice between coal and gas is a question of price, and particularly if the gas is bought on a sliding-scale basis, its cost going down as the load, and consequently the demand for gas, go up. Continuity of the gas supply is another factor, but it is not difficult to arrange the furnaces so that the change from one fuel to the other can be quickly made.

EFFECT OF NONCOMBUSTIBLE CONTENT OF COAL

The subject of the "Effect of Noncombustible Content of Coal" was opened by Thomas B. Coghlan, of the American Gas and Electric Co., Canton, Ohio. Briefly, he showed what effect the noncombustible content of the coal has on the efficiency of boilers. If the coal is not cut clean in the mine and the fire-clay and slate not separated, the ash content will be 50 to 60 per cent. greater than need be. The operator gets it going and coming, or in other words, there is more fuel coming in and more ashes to remove. Besides, there is an increasing loss of combustible as the ash content increases, and it is necessary to burn the coal closer to the ash. The big factor in burning coal high in ash is the loss of capacity, which requires the operation of additional units or the forcing of an existing installation with perhaps an added loss in efficiency.

TABLE 1. EFFECT OF INCREASED ASH CONTENT IN COAL

Ash in Coal, per Cent.	Combustible in Coal, per Cent.	Combustible Loss in Ash, per Cent.	Equivalent Coal Loss, per Cent.	Available Combustible, per Cent.	Evaporation per Lb. Coal	Efficiency of Boiler and Furnace	Capacity, per Cent.	Ash and Refuse, per Cent.
7	87	3.00	3.45	84.0	9.76	76.0	100	10.45
8	86	3.42	4.00	82.6	9.57	75.5	98	11.42
9	85	3.85	4.53	81.1	9.42	74.9	96	12.85
10	84	4.28	5.09	79.7	9.26	74.4	95	14.28
11	83	4.71	5.67	78.3	9.09	73.7	93	15.71
12	82	5.14	6.26	76.8	8.92	73.2	91	17.14
13	81	5.56	6.86	75.4	8.76	72.6	90	18.56
14	80	5.99	7.48	74.0	8.58	72.1	88	19.99
15	79	6.42	8.12	72.6	8.43	71.6	86	21.42
16	78	6.84	8.77	71.2	8.27	71.0	84	22.84
17	77	7.27	9.44	69.7	8.05	70.4	82	24.27
18	76	7.70	10.13	68.3	7.93	69.8	81	25.70
19	75	8.13	10.84	66.9	7.77	69.2	78	27.13
20	74	8.56	11.57	65.4	7.59	68.6	76	28.56

The accompanying Tables 1 and 2 were presented, the former showing how increased ash content decreased the available combustible and increased the ash and refuse. The table is based on 6 per cent. moisture in the coal and on the ash and refuse containing 30 per cent. combustible. Table 2 shows the increase in loss of combustible as the ash content of the coal increases.

CO-OPERATION BETWEEN SMOKE PRODUCERS

At luncheon Robert F. Dennison, of the Cleveland Chamber of Commerce, gave an interesting address on "Coöperation Between Smoke Producers." After reviewing the work of the smoke-prevention department since its inception in 1903 and its failure to produce results first through the lack of power and later through delays in the conviction of violators and the remission of fines after the courts had passed sentence, the speaker announced that Cleveland is to try out a new method in which coöperation rather than force will be the watchword. The thing to do is to convince the smoke producer that smoke is not necessary and that higher efficiency can be obtained without it. A voluntary organization of fuel burners such as has been established in Hamburg, Germany, is to be formed.

The German association not only seeks to reduce smoke, but through tests and periodical inspections attempts to improve the economy of the plant. The Cleveland Chamber of Commerce has already appropriated sufficient money to make tests on a couple of plants that have been violators. If the results of the tests show that these plants can be operated without smoke at higher efficiency, then it is proposed to organize a voluntary association which will give to the small plant the same kind of service the large plant can demand from its experts and efficiency engineers.

TABLE 2. COMBUSTIBLE LOSS IN ASH FOR VARYING PER CENT. OF ASH IN COAL

Per Cent of Ash in Dry Coal	Per Cent. of Combustible in Ash and Refuse								
	10	15	20	25	30	35	40	45	50
	Per Cent. of Combustible Loss								
7	0.77	1.23	1.75	2.33	3.00	3.76	4.62	5.72	7.00
8	0.88	1.41	2.00	2.66	3.43	4.30	5.28	6.54	8.00
9	0.99	1.58	2.25	2.99	3.85	4.84	5.94	7.36	9.00
10	1.10	1.76	2.50	3.33	3.28	5.38	6.60	8.18	10.00
11	1.21	1.95	2.75	3.66	4.70	5.91	7.26	8.99	11.00
12	1.32	2.11	3.00	3.99	5.13	6.45	7.82	9.81	12.00
13	1.43	2.29	3.25	4.32	5.56	6.99	8.58	10.63	13.00
14	1.54	2.47	3.50	4.65	5.99	7.58	9.24	11.45	14.00
15	1.65	2.64	3.75	5.00	6.42	8.07	9.90	12.27	15.00
16	1.76	2.86	4.00	5.33	6.85	8.61	10.56	13.00	16.00
17	1.87	2.99	4.25	5.66	7.27	9.14	11.22	13.80	17.00
18	1.98	3.17	4.50	5.99	7.70	9.73	11.88	14.72	18.00
19	2.09	3.35	4.75	6.32	8.13	10.37	12.54	15.54	19.00
20	2.20	3.52	5.00	6.66	8.56	10.76	13.20	16.36	20.00

To get started, either stock will be issued and sold or assessments levied on members. It is then proposed to charge a certain rate per ton of coal for services rendered. If it is possible to save 10 to 20 per cent. of the fuel formerly burned, as at Hamburg, then a charge of 5c. per ton would not be exorbitant.

The acting force of the association will at first consist of a chief engineer, an assistant, an expert fireman and a stenographer. It is proposed to give the plant of each member a standard test and analysis. Improvements will be suggested and the services of the expert fireman tendered for a time.

There will be an exchange of data between plants, and two or three times a year the fuel for each plant will be analyzed. Osborn Monnett, formerly smoke inspector of Chicago, and now with the American Radiator Co., will head the committee that will conduct the first trial tests.

Simple Test of Coal

There is a simple form of test for coal that ought to be known to everyone interested in fuel. It is based on the quantity of pure lead that will be released from its oxide by a given weight of carbon. Litharge is oxide of lead and contains 34.5 units of lead to one of oxygen. When finely powdered coal mixed with oxide of lead is heated to the combining temperature, the oxygen of the compound unites with the carbon and leaves the metallic lead. The quantity of lead precipitated gives the means of determining how much carbon there was in the coal used.

Let a sample of coal be heated with 40 times its weight in pure litharge. The weight of the former will give a basis for ascertaining the purity of the coal. The weight of lead will vary from 20 to 30 times the weight of the coal. If the weight of the lead is 25 times the weight of the coal employed, the percentage of carbon in the coal will be about 25 ÷ 34.5. This is not absolutely correct, but it is near enough for all practical purposes.—"Railway and Locomotive Engineering."

Measurement of Steam Flow*

By E. G. Bailey†

Few problems have been more baffling to the mechanical engineer than the accurate metering of steam in large quantities. It is necessary to use the "flow" principle; that is, to produce a pressure difference that varies in a definite and known relation to the rate of flow and then to accurately measure this pressure difference in terms of weight of steam.

There are only two fundamental principles that have been used to obtain the pressure difference—"impact" and "change of velocity." The "impact" principle is used in the pitot tube with its various modifications; the vortex[1] and metering bend[2] also belong to this class. The "change in velocity" type covers the venturi tube and the orifice.

We do not usually think of the orifice as being of the same basic principle as the venturi, but the two are practically identical when the orifice is used in a pipe and the pressure connections are made at the proper place. The pressure difference produced in both cases is caused by the increased velocity and the corresponding change in the static pressure between two different positions along the pipe where the flowing stream has different areas. The exact area and location of the smallest section of the stream on the downstream side of an orifice may not be quite as tangible a thing to see and measure as is the throat of a venturi tube, to which it corresponds, but experiments have proven that it is none the less a perfectly dependable function of the opening in the orifice plate. With high initial velocities, the pressure difference is largely restored in static pressure within four pipe diameters beyond the orifice so that the net loss in head caused by the orifice may not be more than 25 per cent. of the difference in pressure available. This compares very favorably with the best design of venturi tube where the net loss is about 15 per cent. of the total venturi head.

Pitot tube and multi-hole nozzles of various design and modification were used exclusively during the first two years of experiment, but were abandoned for reasons explained later. The venturi tube was at first considered to be the only available substitute, but the expense of changing large steam pipes, in addition to the cost of the tube itself made this prohibitive for general use. The orifice was taken up and the following important points were considered in working out a suitable design: (1) The orifice plate must go in between a pair of existing pipe flanges; (2) it should not thicken the joint so as to throw them out of parallel; (3) the metal used should withstand corrosion and abrasion; (4) it should be capable of easy installation.

THE ORIFICE PLATE

The initial design consisted of a $\frac{1}{32}$-in. monel metal plate[3] having corrugations on its outer edge, to make its own packing the same as a corrugated copper gasket, and having a round concentric hole smaller than the pipe diameter. But what coefficient to allow in calculating the size of the orifice for a given pipe diameter, steam density and rate of flow was the big unknown. The importance of testing the orifice to see if it would maintain a constant factor over long periods of time was of more importance than to know the exact factors under all conditions, consequently some guesses as to the coefficient were made and several orifices were put into experimental use in places where the service was continuous and severe.

One of the first orifices of this type was put into a 15-in. pipe supplying steam at 200 lb. pressure and 150-deg. superheat to a 15,000-kw. turbine in the L Street Station of the Edison Electric Illuminating Co., of Boston, where the rate of flow averages more than 150,000 pounds of steam per hour. The only flange available was at the inlet to the turbine-strainer, about 6 ft., or five pipe diameters, beyond a 90-deg. bend of 6 ft. radius. The testing department of the Edison company has made weighed water-rate tests of this turbine each month since this orifice was installed in December, 1913, and results from this orifice have been obtained at the same time.

The substance of these tests is given in Table 1. There are 28 tests extending over more than two years, and approximately 1,000,000 tons of steam has passed through this orifice in this interval of time. It is noted that the greatest difference is 2.51 per cent. and the majority of them are within 1 per cent. The device for measuring the pressure difference in this case was not of the latest and best design.

It is shown conclusively that there is no appreciable change in the orifice factor over this period. This is also borne out by the measurement and appearance of orifice plates replaced by others of larger capacity after a year or more of service, none of which show any corrosion or wear whatever.

Other orifices were put in from time to time under various conditions where weighed-water calibration data could be obtained at capacities most frequently met in practice. The coefficient of discharge did not appear to be as constant for different ratios of orifice to pipe diameter as it had been hoped that it would. The indications were that the orifice was fundamentally correct, and the discrepancies encountered were only through lack of sufficient knowledge relating to details. The general trend of the data up to this point was that the coefficient of discharge increased with an increase in the ratio of orifice diameter to pipe diameter, with no obvious reason why it should. In cases where the area of the orifice is large compared with the area of the pipe, the velocity of approach is an important factor, but even taking this into consideration, the net coefficient of discharge increased with the orifice area. This same fact has since been brought out by Hickstein[4], who practically followed the results obtained in Charlottenburg[5] some years ago. These two experiments dealt with natural gas and air only, and unfortunately the pressure connections were located at such great distances from the orifice plate that they could be duplicated in but very few steam pipes where there were long straight runs.

LOCATION OF PRESSURE CONNECTIONS

The question that had never been thoroughly investigated to our knowledge was the location of the pressure connections on the upstream and downstream sides of the orifice. It was felt that this was a point that should be thoroughly investigated, consequently at the writer's suggestion, Professor Judd, of Ohio State University, started some experimental work along this line, and some of his results were presented in a paper at the New Orleans meeting of this year under the title of "Experiments on Water Flow Through Pipe Orifices."

The important points brought out by Mr. Judd's data are:

1. The coefficients of the orifices tested are substantially constant at all rates of flow for any one set of pressure connections regardless of their location with respect to the orifice plate.

2. The location of the pressure connections is of vital importance with the higher ratios of orifice to pipe diameter with round concentric orifices, due to the restoration of static pressure.

3. The use of specially designed orifices of other shapes than round and concentric, permit the downstream connection to be far enough away to clear any flange and also allow more latitude in the location of this connection without endangering the accuracy.

DISTRIBUTION OF PRESSURE DIFFERENCE

The last two points are clearly illustrated in Fig. 1, which shows a diagrammatic arrangement of the experiment and the plotted curve representing the pressure difference as measured between the upstream connections and the several downstream connections located at ½-in. intervals from the orifice plate. It is evident from the round concentric orifice of relatively large diameter in the upper view of this illustration that the action of the fluid on the downstream side of the orifice is practically identical with the venturi tube. The point of highest velocity is at the "vena contracta," just beyond the orifice plate, and the static pressure is the least and the pressure difference the greatest at this point, which corresponds to the throat of the venturi tube. The active stream expands beyond this point, and as its velocity is thereby reduced, the static pressure is partly restored and the pressure difference correspondingly diminished. As high as 77 per cent. of the pressure difference is restored in some cases.

It seems that the orifice acts merely as a forming tool to direct the active stream into a natural venturi shape, and at the same time gives just as dependable results. In fact, it appears to give more dependable readings than a venturi

*Abstract of a paper presented before the Boston Section of the A. S. M. E., May 23, 1916.

†President of the Bailey Meter Co.

[1]Hayes, "A. S. M. E. Trans.," Vol. 36, 1914.

[2]Levine, "A. S. M. E. Trans.," Vol. 36, 1914.

[3]"Power," May 9, 1916, p. 672.

[4]"The Flow of Air Through Thin-Plate Orifices," Annual Meeting, A. S. M. E., December, 1915.

[5]"Zeit. des Ver. d. Ing.," Feb. 22, 1908.

tube when placed close beyond or in front of bends or other disturbances. Similar orifices put in under the following conditions gave the same coefficient within 1 per cent. when calibrated by actual weight tests:

Reference No.	Size of Pipe, In.	Measuring	Straight Run on Upstream Side Pipe Diam.	Conditions on Upstream Side of Straight	Straight Run on Downstream Side, Pipe Diam.	Conditions on Downstream Side Straight
1	15	Steam	5	90 deg. bend	None	Turbine strainer
2	8	Steam	1½	90 deg. bend	None	Turbine strainer
3	6	Steam	3	90 deg. bend	None	Gate valve
4	6	Steam	20	Gate and globe valve	20	Orifice
5	5	Water	59	Elbow	56	Orifice
6	4	Steam	6	Outlet of 4-in. tee	none	Gate valve

The writer believes that a helical or corkscrew motion of the fluid following a bend or similar disturbance is very common and prevents any satisfactory results being obtained from the use of pitot tubes, multi-hole nozzles or venturi tubes, while the orifice seems to give the same coefficient as if in a long straight run of pipe. To illustrate this the following interesting experience is given:

It was desired to measure the steam output from some 250-hp. boilers having 8-in. steam outlets which gave a very low steam velocity. The boilers were standard 2-drum B. & W. with the 8-in. pipe rising from the manifold with a 90-deg. bend and then about 3 ft. of straight horizontal run, into a gate valve adjacent to the header. At first a multi-hole nozzle similar to that shown in Fig. 2 was located about a foot in front of the gate valve, and upon calibration it was found that

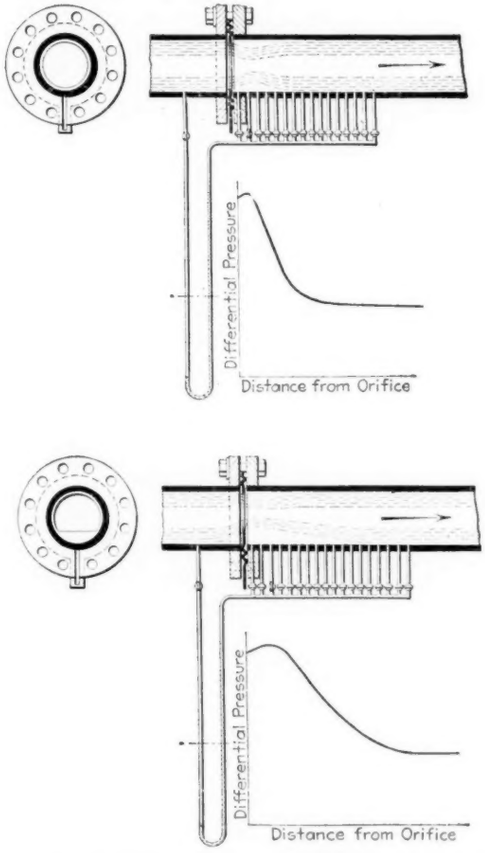


FIG. 1. DISTRIBUTION OF PRESSURE DROP THROUGH CONCENTRIC AND ECCENTRIC NOZZLE

the reading was about 15 per cent. lower than would be expected. At first it was assumed that the highest velocity was not in the center of the pipe, but was shooting above the nozzle as a result of the 90-deg. bend. Traverse work in the steam pipes from the boilers was not attempted, owing to the fluctuating rate of steam flow and the difficulty of handling adjustable tubes in high-pressure steam lines. A single-hole nozzle like that shown in Fig. 3 was then tried. A rather unexpected set of results was obtained, for during the nighttime, when the load on the boiler was very light, it was found that this nozzle gave very nearly the same factor as applies to a long straight run of pipe, but as soon as the load came on in the morning, instead of the steam-flow pen starting up, it would drop. If the load was sufficiently increased, it would come back to the same reading given on the night load or a little higher, but never did it go anywhere

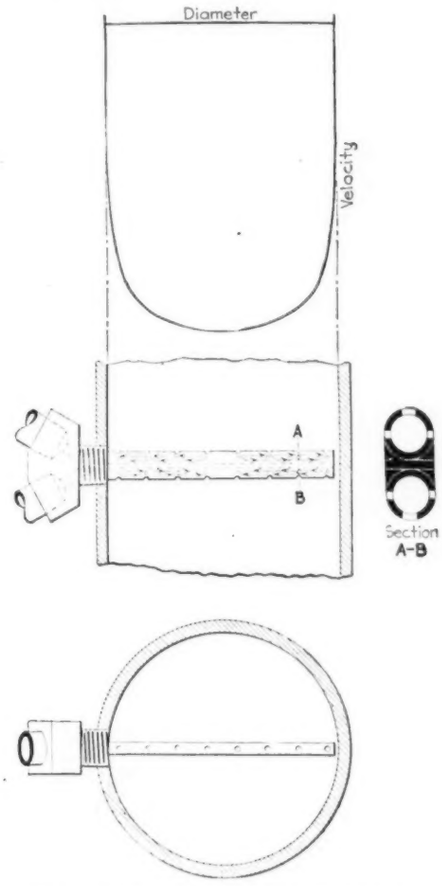


FIG. 2. PLAN, ELEVATION AND SECTION OF MULTI-HOLE NOZZLE WHICH INDICATED SWIRLING

near as high as it should. Some calibration work under the heavy-load conditions showed that this single-hole nozzle gave results as much as 85 per cent. low.

In hunting for an explanation all available literature on the subject was searched and the paper of Gregory and Schoder[6] gave practically the only data pertaining to traverses of pipe lines in and near a bend. Some very peculiar flow curves were shown in this paper with practically no comment, but by plotting these flow curves on pieces of paper and pasting them at right angles to each other as the traverses were made and stringing them on a bent wire, it was very evident that a swirling motion was taking place, which resulted in a higher velocity near the circumference of the pipe than existed in the center. This coincided with our experience, because the multi-hole nozzle showed the velocity in the center of the pipe to be normal at low velocities, but very much below normal at velocities that might be expected to produce the helical motion.

To prove this point beyond doubt, the next step was to put in a single-hole nozzle, or the equivalent to a pitot tube, with the openings near the circumference of the pipe. This was done, and it was found that a very high velocity existed within ½ in. of the circumference of the pipe when the boiler was steaming at 50 per cent. or more of its rated capacity.

This experience killed all faith in the pitot tube and any other type of nozzle for use in steam pipes where there was any possibility of the helical or swirling motion. Orifices were installed in these boiler connections at the inlet to the gate valves with very accurate and satisfactory results.

The gradual contraction at the inlet to a venturi tube would have no tendency to destroy the helical motion, and its angular velocity would be greatly accelerated in the throat, thereby producing a centrifugal force against the piezometer holes

[6] "Trans. A. S. M. E.," Vol. 30, 1908.

and give an erroneous pressure difference. The more sudden contraction in the orifice and its downstream-pressure connection, being entirely away from the active stream, are undoubtedly responsible for the satisfactory results from its use.

In addition to the fact that the multi-hole nozzle did not give a true average in the instance referred to with the helical motion of steam, it failed to maintain a constant factor even in long runs of straight pipe. The type of nozzle used in these experiments is shown in Fig. 2. This was made by driving two pieces of $\frac{5}{16}$-in. phosphor-bronze tubing into a plug having a $\frac{3}{4}$-in. pipe thread to be screwed into the side of the steam pipe. Regardless of the size of pipe, the upstream tube was drilled with eight $\frac{3}{32}$-in. holes equally spaced across the diameter of the pipe. The downstream tube was drilled in the same manner, so that it got the reverse

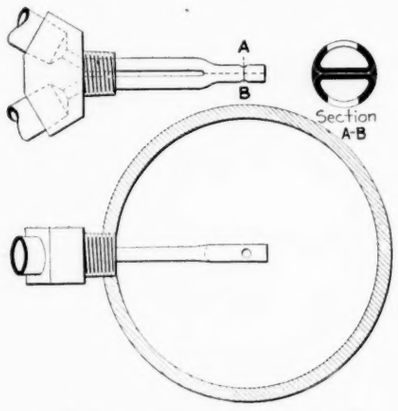


FIG. 3. SINGLE-HOLE NOZZLE

effect instead of being a true static. The side-outlet holes shown were used only for high velocities. The theory of this multi-hole nozzle is not based upon very sound premises, for in the case of a normal flow in a long run of straight pipe with the higher velocity in the center, it is obvious that the dynamic force against the holes near the center of the pipe will be greater than that against the holes near the side, therefore steam will enter the center holes, divide and run both directions through the $\frac{5}{16}$-in. tube and be forced out against the flowing stream through the holes near the side of the pipe. The net result in so far as pressure or pressure difference is concerned will therefore be a function of the relative area of these holes as compared with the area and length of the tube in which they are drilled, together with the spacing of the holes along the tube with respect to the diameter of the pipe. Considerable experimenting was done with this type of nozzle in various sizes of pipes, and its coefficient was found to vary between $\frac{1}{19}$ and $\frac{1}{21}$.

Early in our experience some very high velocities were encountered in the case of 500-hp. boilers provided with 5-in. steam pipes, as the boilers were frequently operated at 220 per cent. or more of their rated capacity. The velocity head that would be produced by the multi-hole nozzle, as shown in Fig. 2 with only the upstream and downstream holes would have exceeded 7 in. mercury, which was beyond the capacity of the recording device being used. It was therefore necessary to minimize this pressure difference in some way. It was first decided to drill bypass holes between the upstream and downstream tubes of the nozzle, but this was never tried, because the tubes were not always exactly the same distance apart and in drilling through the spacer and brazing material that filled the space between the tubes, it would give the equivalent of a long tube of small diameter with variable and undetermined length. Therefore the holes were drilled in the sides of both the upstream and downstream tubes, which would be top and bottom in a horizontal pipe. Such a tube is shown in Fig. 2. The sizes of these outlet holes were varied according to the velocity to be contended with and the pressure difference desired.

A large number of calibrations were made with such multi-hole nozzles in 5-, 6- and 8-in. pipes, with outlet holes varying from a No. 60 drill up to a No. 47 drill, in which case the coefficient was brought down as low as 0.68, as against 1.2 with no side-outlet holes.

Continued experiments with these multi-hole nozzles showed their vital weakness to be in a change in coefficient resulting from the accumulation of sediment around the edges of some of the holes, thus reducing their area. In many cases some of the holes were entirely plugged. In some steam pipes this sediment accumulated very rapidly, so that no dependable results could be obtained over any reasonable length of time. Even the simpler type of nozzle where there were no outlet hols, and only these pointing directly upstream and downstream, produced the same trouble, for sediment would accumulate around the holes at the radiator end of these tubes and a higher pressure difference was produced, owing to the relatively greater effect of the holes near the center of the pipe. In time the center holes would become partly or completely plugged, in which case the coefficient would again approach normal, and then it would decrease when only the holes in the low-velocity zone at the farther side of the pipe were left open. In the latter case the meter would read low. Larger holes minimized or delayed the change in the factor, but they did not prevent it.

The question of measuring steam to reciprocating engines, pumps, etc., has been somewhat misunderstood. Many engineers have the impression that the pulsations cause an unduly fluctuating motion of the recording or integrating mechanism, so that accurate results cannot be secured; but such is not the case except in very slow-speed pumps. It is also wrong to say that a flow meter is inaccurate for measuring pulsating flow, for the present flow meters will produce accurate results when more knowledge is gained along this line. Steam flowing to a reciprocating engine follows some such variations as shown in Fig. 4, the exact nature of curve depending upon the cutoff, capacity of steam line between the orifice and the engine, and some other factors. At any rate of flow there are great and frequent variations in the rate of flow, which occur periodically and yet so rapidly that no recording meter could possibly follow them individually. In fact, the inertia of the water columns in the connecting pipes to the recorder, together with the mercury in the U-tube, bell float or other mechanism is so great that no appreciable motion results from the individual pulsations, but rather the reading is an average of the head or pressure difference resulting from such flow, regardless of the use of cams, special-shaped bells, displacing members or any other mechanism whereby a motion is produced in direct proportion to the rate of flow. For continuous flow they cannot possibly produce an average reading that is directly proportional to the rate of flow in the case of pulsating flow.

In Fig. 4 the parabola shows the characteristic relation between rate of flow and pressure difference with 100 per cent. flow at 100 per cent. flow head. Assuming that the rate of flow varies from zero to 100 per cent., the nature of the cycle being no flow during 50 per cent. of the time and 100 per cent. flow during the other 50 per cent. of the time, then the average flow would be 50 per cent. At the same time the pressure difference varies in like manner from 0 to 50 in. water head with an average of 25 in. The inertia of the meter mechanism produces a reading corresponding to this average pressure difference of 25 in. instead of an average reading of 50 per cent. on the quantity scale. The reading on the quantity scale corresponding to the average head of 25 in. is 70.7 per cent. This 70.7 per cent. divided by 50 per cent., which is the true average flow, gives 1.42, or a reading 42 per cent. higher than would be the case if the same amount of steam

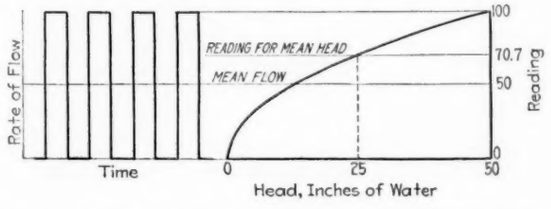


FIG. 4. SHOWING EFFECT OF FLUCTUATING FLOW

were flowing at a continuous rate. This ratio has been termed the "pulsating factor," and while it might be calculated on a theoretical basis from the nature of the flow curve if it were known, yet it is better to determine it by actual calibration and if so determined, the steam-flow meter can operate as satisfactorily on a reciprocating engine as on continuous flow to turbines, heating systems, etc. It should be remembered that the pulsating factor may not be the same at all different loads of the engine. It is, however, a function of the load for any one engine so long as the valve adjustment, etc., remain reasonably uniform.

It is desirable to have a ready means of calibrating steam-flow meters in place, the same as it is to calibrate wattmeters, pressure gages, thermometers, etc. It has heretofore been considered necessary that steam-flow meters be calibrated by passing all the steam through a turbine to a surface condenser that is known to be tight and free from leaks. This latter point can be well taken care of where sea water is used

for circulating, but is often difficult to prove where fresh water is used.

The other method is by weighing water to a boiler from which all the steam passes through the meter being calibrated. In this case it is necessary to guard against leaks from the feed line, blowoff or branch steam lines. An additional difficulty is in having the same amount of water in the boiler at the end as in the beginning of the test. The fact that the water level is brought to the same mark on the gage-glass is not positive proof that the water in both drums or even different parts of the same drum is at the same level. But a still greater source of error along this line lies in the volume of steam beneath the water level, which is greatly affected by the rate of steaming. Some tests on a 500-hp. B. & W. boiler have shown that the volume of steam beneath the water level varies as much as 50 cu.ft. with a change in rate of steaming of 14,000 lb. per hour. That is, the boiler was steaming at the rate of 24,000 lb. per hour at the beginning of the hour and at 10,000 lb. per hour at the end of the hour; there were 11,500 lb. of steam generated during the hour, but 14,750 lb. of water was required to be fed to the boiler during this period to maintain the water level at the same point in the gage-glass. This means that 3,250 lb. of water, or over 50 cu.ft., was required to fill the additional space occupied by steam below the water level at the higher rate of steaming, as compared with the lower rate. This is a source of error that cannot be overcome or corrected except by continuing a test over a long period of time and ending the test with the same rate of steaming as at the beginning.

In case of weighing water either to a boiler or from a surface condenser, it is necessary to put in large weighing tanks and scales and often makes very costly changes in the feed line or steam piping, so that a diligent study of this

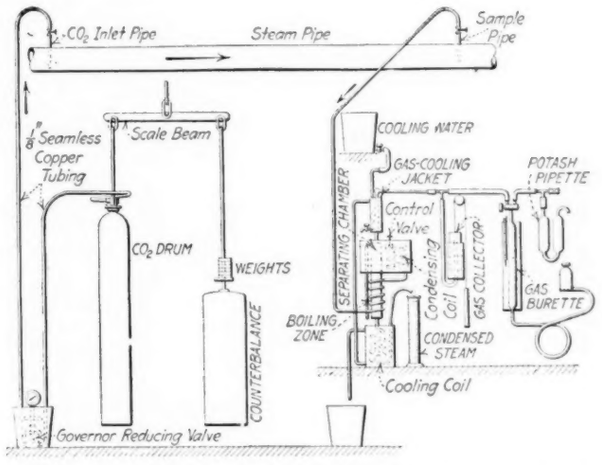


FIG. 5. CO_2 METHOD OF MEASURING STEAM FLOW

problem was made to find a better method of steam-meter calibration. The method worked out by the writer consists in introducing CO_2 gas from a high-pressure drum into the steam line at a known and continuous rate, permitting it to mix with the steam; then drawing out a sample, condensing the steam and determining the ratio of condensed steam to CO_2 gas.

In the first experimental work along this line the attempt was made to determine this ratio by considering the CO_2 as carbonic acid in solution and titrating for the amount of it contained in the condensed steam by the use of chemical means. Some rather erratic and unexpected results were at first obtained from this titration method, and in an effort to check up this method another more valuable feature was worked out whereby the gas was thoroughly separated continuously from the condensed steam and the two measured separately. The general diagrammatic arrangement of the apparatus used in this method is shown in Fig. 5.

This method is based upon the addition of CO_2 gas from a high-pressure tank or drum to the steam line at a continuous and known rate. The gas becomes thoroughly mixed with the steam as it passes along, and a sample consisting of a mixture of steam and CO_2 gas is drawn off continuously from a point farther down the line. The steam in the sample is then condensed, the gas and water are thoroughly separated and each measured so as to determine the ratio of water to gas. This ratio multiplied by the rate at which the gas was added will give the rate of flow of steam.

The drum containing CO_2 gas under high pressure is supported on a scale beam so that its weight can be determined at any time, from which the rate of adding the gas is accurately determined. The gas is continuously discharged through a flexible copper tube and a governor valve into the steam line through a ⅛-in. spray tube. A sample of the mixed steam and gas is taken from the steam pipe at a point farther down the line. The sample is then passed through a condensing and separating apparatus, and the condensed steam and gas are measured separately in a graduate and Hempel burette respectively. The entire operation is continuous except that these two measurements are taken simultaneously at regular intervals of four to five minutes.

TABLE 1. WEIGHED WATER RATE TESTS

Results from a steam-flow meter, orifice type, measuring steam to a 15,000-kw. turbine in L Street Station of the Edison Electric Illuminating Company of Boston. This meter has a maximum capacity of over 200,000 lb. per hr. and is provided with a monel metal orifice placed in the inlet flange of the turbine strainer about 6 ft. beyond a 90-deg. bend in a 15-in. pipe. Meter readings were taken in conjunction with the regular monthly turbine water-rate tests by their testing department.

Date	Duration of Test	Steam Pressure by Gage, Lb.	Super-Heat, Deg. F.	Condensed Steam per Hour As Weighed, Lb.	Steam per Hr. by Orifice Meter, Lb.	Difference Between Meter and Weight, %
Jan. 8, 1914	1 hr.	196.5	129.6	133,417	134,372	+0.72
Feb. 6, 1914	1 hr.	191.3	152.2	137,739	138,756	+0.74
Mar. 28, 1914	55 min.	200.1	128.4	136,763	138,163	—0.44
Apr. 23, 1914	1 hr.	190.7	135.4	136,332	136,366	+0.02
May 14, 1914	1 hr.	193.1	149.0	132,610	134,299	+1.27
June 12, 1914	35 min.	196.5	134.1	135,421	138,495	+2.27
July 9, 1914	1 hr.	191.3	128.8	138,740	141,300	+1.85
Aug. 15, 1914	1 hr.	194.2	130.8	142,065	143,235	+0.84
Sept. 10, 1914	1 hr.	197.0	129.3	136,846	138,606	+1.29
Oct. 20, 1914	1 hr.	197.5	138.2	137,003	139,723	+1.99
Nov. 21, 1914	1 hr.	193.9	140.8	132,552	135,631	+2.32
Dec. 19, 1914	1 hr.	196.0	146.2	129,759	130,454	+0.54
Jan. 22, 1915	1 hr.	198.0	152.1	131,134	127,845	—2.51
Feb. 19, 1915	1 hr.	195.2	163.7	129,165	127,641	—1.18
Mar. 18, 1915	1 hr.	193.0	168.3	128,917	126,317	—2.02
Apr. 24, 1915	40 min.	195.0	158.5	132,017	129,856	—1.56
May 20, 1915	1 hr.	197.4	158.6	132,107	134,120	+1.52
June 22, 1915	1 hr.	193.5	145.8	140,582	138,340	—1.59
July 20, 1915	1 hr.	192.8	141.2	138,457	137,110	—0.97
Aug. 19, 1915	1 hr.	192.6	141.5	139,200	138,975	—0.16
Sept. 23, 1915	1 hr.	196.8	143.0	139,385	137,399	—1.42
Oct. 27, 1915	1 hr.	195.0	142.3	137,504	135,472	—1.48
Nov. 19, 1915	1 hr.	193.8	152.1	137,397	134,528	—2.09
Dec. 7, 1915	1 hr.	196.1	154.2	131,466	129,205	—1.72
Jan. 18, 1916	1 hr.	196.0	152.2	130,526	130,610	+0.06
Feb. 11, 1916	1 hr.	194.1	141.3	131,812	132,001	+0.14
Mar. 14, 1916	1 hr.	195.2	171.4	129,030	130,391	+1.05
Apr. 13, 1916	80 min.	189.4	144.2	137,472	136,945	—0.38

An accuracy of 1 to 1½ per cent. can be secured when using only 1 lb. of gas to 2,000 lb. of steam. Hence it is possible to use this method on large capacities up to several hundred thousand pounds of steam per hour. Some data are given in Table 2.

This method has also been tested out for measuring the flow of water, the gas being injected into the feed line, and with the small ratio of gas required the CO_2 is completely absorbed by the water, so that a representative sample is readily obtained. The gas is then completely separated from the water and the ratio determined by the measurement of each.

It will be noticed that this method is entirely independent of steam pressure, superheat or percentage of moisture and it resolves back to an actual weight basis. It requires no change in piping except to drill and tap for ⅛-in. pipe at two places. It is particularly adapted for test work in calibrating meters determining steam consumption of turbines, engines, pumps, auxiliaries, etc., but does not take the place of a permanently installed recording or integrating meter.

[The speaker, by request, described briefly the boiler indicating apparatus shown on pages 807 et seq. of "Power" for June 6.—Editor.]

DISCUSSION

WALTER S. BURKE, of Harvard College, told of the use of Bailey meters in measuring the steam supplied to the college by the Boston Elevated Railway Co., which installation and use were described in a paper by Mr. Bailey at the recent meeting of the National District Heating Association. The CO_2 method of calibrating the meters in place is perhaps one of the most valuable features in connection with the apparatus. He emphasized the fallacy of attempting to measure the evaporation of a boiler by noting the difference in the water level on account of the varying amounts of steam in the water from time to time. It is not possible to tell at any time what the rate of steaming is by the rate of feeding. They have at Harvard two meters which they can place in series, and they check each other very closely, their accuracy, as determined by measuring the condensation with a weir meter, being within one-tenth of 1 per cent.

ALBERT G. DUNCAN, of the Harmony Mills, said that the Bailey meters which they had in use checked very closely with the venturi meter on the feed-water line.

A very considerable saving can be made by watching the steam actually going into the heating system as compared with the outside temperature, and this can be done only by separating the heating steam from that used in other processes, and metering it. The saving amounts to some 25 per cent. over the old rule-of-thumb method.

F. F. UEHLING, of the Uehling Instrument Co., said that the measurement of steam from the boiler is undoubtedly of great importance and has many fields of application. On the other hand, the assumption of a definite amount of air supplied for a definite amount of steam produced is radically wrong, owing to the fact that fuel varies physically and chemically. The barometric conditions and humidity all have an effect on the amount of air required to burn a given amount of coal. Mr. Bailey no doubt uses the CO_2 determination to fix the relation of the air supply to the steam produced. The percentage of CO_2 in the products of combustion is an actual measure of the volume of the products of combustion per pound of coal used, therefore the greater percentage of CO_2 the less products of combustion per pound of coal used, and less air to be heated and sent away, and this certainly seems to be a very simple method of getting at the loss directly. Mr. Bailey had referred to the difficulties that he had had with different sorts of orifices and the nicety that is necessary in their proportioning and connection, but in his boiler apparatus he uses the tubes or gas passages of the boilers as an orifice for the air, and these, owing to fouling, etc., must be a very unsatisfactory sort of an orifice, with all sorts of opportunity for eddy currents, etc.

JOHN A. STEVENS, consulting engineer, told of some very satisfactory results that he had obtained with pitot tubes. He had checked them with measuring tanks, and had come within 5 per cent., sometimes as low as 2 per cent. They have used one meter connected to nine different sources of supply, and the results of the different indications varied less than 1 per cent.

TABLE 2. COMPARATIVE SUMMARY

Summary of data comparing steam measurement by CO_2 method with actual weight and steam-flow meter, orifice type. Made in connection with evaporation tests on 512-hp B. & W. Boiler.

Run No.	Duration of Test, Hr.	Water Fed to Boiler by Actual Weight, Lb.	CO_2 Fed per Hr. Lb.	Ratio Water to CO_2	Steam by CO_2 Lb.	Steam by Meter, Lb.	Difference Between CO_2 Method and Actual Weight, %	Difference Between CO_2 Method and Steam Meter, %
1	4	.....	8.64	2,548	90,500	91,620	.....	—1.22
2	2.5	78,515	8.53	3,645	77,650	78,040	—1.10	—0.50
3	6	133,474	8.68	2,506	130,500	129,190	—2.26	+1.01
4	3	54,505	14.63	1,276	56,000	55,830	+2.78	+0.29
3 and 4	9	187,979	.....	.....	186,500	185,020	—0.78	+0.80
5	5	132,667	8.85	2,992	132,130	133,650	—0.40	—1.14

A. R. DODGE, of the General Electric Co., expressed his appreciation of the paper, but did not discuss it in detail.

W. S. BURKE asked if there were any satisfactory method of measuring water fed to a boiler after it had left the feed pump, to which Mr. Bailey replied that he had recommended a great many venturi meters for that purpose. Its indications were susceptible to distortion from pulsating flow, but that difficulty is not confined to any one type of meter. In feed water the inertia of the water is harder to deal with than that of steam, and he would prefer to measure steam than water.

CHARLES T. MAIN said that twenty-five years ago or so manufacturers were making a great variety of guesses as to the cost of their production; some goods they knew they made at a profit, but others they were not sure that they did not make at a loss. Steam is a very considerable item in the cost of production, and the use of a steam meter enables them now to determine this item with precision.

SANFORD A. MOSS, of the General Electric Co., doubted the accuracy to be obtained by the use of the tube passages as an orifice. He said that the tubes might be clogged up, and in the last analysis he thought that if we were to get the right connection between the air and steam by means of CO_2 analysis, why not keep on using the CO_2 instead of using the air at all?

ROBERT E. DILLON, of the Edison Electric Illuminating Co., of Boston, said that a few years ago an engineer in using the differential pressure between the two sides of an orifice to measure the amount of fluid passing through a pipe found it necessary to calibrate the orifice in the position and under the circumstances in which he was to use it. Many different coefficients of discharge were found under the various conditions of use, and this was usually attributed to the fact that the fluid had just passed an elbow, valve or other obstruction, which had caused an abnormal condition of flow in the pipe.

Considerable progress has been made in recent years in determining what the coefficient of discharge is under various conditions. An interesting paper on this subject is the junior prize paper of the association. The author has done much to establish the coefficient under all conditions where the differential pressure is measured between two points a considerable distance upstream and downstream from the orifice. This coefficient apparently only changes with different percentage of orifice to pipe diameter. Consequently, knowing this ratio, the flow of gases or liquids can be measured by an orifice with considerable accuracy without the necessity of calibration. There is, however, a disadvantage to this method due to the fact that the differential pressure at remote points is relatively small, necessitating a comparatively small orifice to get proper pressure differences, and consequently a relatively large loss of pressure in the pipe line.

Professor Judd, in his paper before the society at the April meeting, describes the investigation which he has carried on to establish the coefficient of discharge as figured from pressure differences taken nearer the orifice. Taking the pressure difference at this position has the advantage in that, for the same pressure difference as measured, the final loss in the pipe line is considerably less.

Professor Judd has established the coefficient for differential pressures taken between a fixed point upstream from the orifice and all points downstream from the orifice, also, between a fixed point downstream from the orifice and all points upstream from the orifice, although he has only given in his paper the coefficient for differential pressures between two fixed points upstream and two fixed points downstream. This work has made it possible to use the orifice for measurements of flow under many more conditions than previously.

Now, there still remains for the rest of us several points to clear up by means of further tests. Tests should be made with all ratios of orifice diameters to pipe diameters and with differential pressures between all points upstream from the orifice and all points downstream from the orifice. With this information at hand it will be possible to insert an orifice, make the pressure connections at the most accessible points and obtain the flow of gas or liquid without the necessity of calibration.

MR. BURKE asked if the coefficient of the orifice varies with the condition of the steam, as to moisture, superheat, etc., to which Mr. Bailey replied that the coefficient does not change, but account must be taken of the density. With the CO_2 method this does not make any difference. The only thing is to get a true sample, which is very easy with superheated steam, and no more difficult with saturated steam than to get a true sample for the determination of moisture. Mr. Bailey insisted that his conception of air as a fuel held good. The number of B.t.u. per pound of oxygen remains remarkably constant, whether that oxygen is combined with wood or coal or oil. If the coal has an extra 10 per cent. of ash, this is simply a diluent and it does not take oxygen to burn it. The gas analysis is ideal as to bringing out the right relation between the air and the rate of combustion, but in order to make out a complete case the oxygen, hydrogen and carbon monoxide must also be determined. The effort to produce a high percentage of CO_2 may be carried to such an extent that unburned fuel is present, and a less favorable result may be obtained with the higher percentage of CO_2. The relation between the air flow and the water evaporated does tell you that you have not enough air going through to generate that amount of steam economically. Another trouble with the Orsat apparatus, and even the CO_2 recorders, is the difficulty of the ordinary firemen to think backward and to get any practical use out of the information that fifteen minutes ago the conditions were such and such.

With regard to the use of the tubes or passages between them as an orifice, Mr. Bailey said that the kinetic energy due to the velocity of the gases in the boiler is nothing like that in a steam line. The critics of the method might have added the disturbance due to baffles being out of order, which is one of the greatest troubles they encounter, but in the satisfactory operation of a boiler room baffles are not supposed to be allowed to leak, nor tubes to be allowed to become clogged up with soot. If they are kept as they should be kept, the apparatus will be all right using them as orifices, and the adoption of an apparatus which requires them to be kept all right may be a good thing rather than a disadvantage. The percentage change in the resistance does not amount to anything compared with the change in the air flow. If you get your air excess down to within 10 per cent. of what it should be, you would be doing very well. If your air resistance changed 10 per cent., you would be up the flue.

PROF. A. L. WILLISTON said that pitot tubes are used with success under some conditions, and asked what length of pipe and under what conditions can the pitot tube be advantageously used. Mr. Bailey replied that the pitot tube was used with success to measure air and gases flowing at low velocities, as in fan work, mine ventilation, etc. By the use of the bell very minute differences in head could be accurately measured.

Report of Station Operating Committee*

SYNOPSIS—This report consists of four parts: (1) That in relation to boiler operation, by means of a set of indicating and recording instruments; (2) water treatment, and the effect of caustic soda on iron and steel; (3) extraction or bleeder type turbines; (4) oil as fuel, showing results to be obtained and the possibility of future development with this commodity as fuel. The following includes only the first part; the other three will appear in subsequent issues.

Boiler Operation

An attempt will be made to illustrate a method of efficiently carrying load by means of the indications afforded by a set of instruments. Under a condition of varying capacity the efficiency of the unit is supposed to remain substantially constant. If this could be maintained exactly, the solution to general plant efficiency would be at hand. While it is not possible to live up to theoretical conditions in this respect, these conditions afford us a standard of comparison.

By combustion efficiency we mean the degree of success in burning all of the combustible in the coal with the minimum air requirements. The time element does not enter into this consideration, so that it is theoretically a matter of indifference whether a pound of coal is oxidized in a minute or an hour. If the minimum amount of air necessary is used, the efficiency of combustion reaches a maximum. It is not possible to determine continuously the efficiency of the combustion process in the boiler furnace. There are losses due to incomplete combustion of gases and the formation of CO and unburned hydrocarbons; there are losses due to the filtration of air both through the fuel bed and the boiler walls, and there are the losses due to the ashpit.

The gas-analysis recorder affords probably the best means for indicating substantially continuously the condition of the flue gases, but unfortunately good CO_2 can be developed at the expense of the ashpit. It is easy to offset all of the advantages of good CO_2 by carrying too heavy a fire and thereby throwing away a lot of combustible. Many so-called efficiency meters, of which several varieties have been placed on the market, take into consideration only the condition of the gases, either as to volume or temperature, but in no way consider the ashpit loss.

Any direct indication of furnace efficiency must include the latter item, and any device indicating or claiming to indicate the over-all efficiency by virtue of the condition of the gases alone is vitally defective. We believe that it is not possible to make any instrument that will clearly indicate overall efficiency under all variations of rating, fuel quality, type of boiler, type of stoker and the personal equation. It may be safely stated that no two boiler installations are the same, and each one must have its own peculiar conditions.

While this is not a simple problem, it is absolutely fundamental and necessary to the economical operation of power plants. Your committee has been earnestly striving, therefore, to work out a practical operating method by which the average plant engineer can gage the combustion efficiency and keep tab on the firemen. This has been worked out with some care in at least one instance, and others are being developed. The instruments by which this object is to be accomplished are shown herewith, Fig. 1, and described briefly as follows:

In the boiler room, directly in front of the stoker, standing against the opposite wall, is the individual panel, or gage board, on which the instruments belonging to the particular boiler are placed. Beginning at the top, the integrating meter is a part of the steam-flow metering equipment built by the Republic Flow Meters Co. and shows the total number of pounds of steam produced by the boiler. This is strictly an integrating meter and is designed to be read daily, so that the total steam output can be checked against the total coal consumption.

*Presented at the eighth annual convention of the National District Heating Association, New York City, May 16-19, 1916. The members of the committee are: Byron T. Gifford, Grand Rapids, Mich., chairman; F. W. Laas, Cedar Rapids, Iowa; E. C. Rogers, Toledo, Ohio; J. G. Deremer, New York City; R. E. Hagenah, New York City; Joseph Harrington, Chicago, Ill.

The upper left-hand meter is the Republic steam-flow indicator, showing the momentary output of the boiler, and is intended for the use of the firemen in judging the momentary condition of the boiler and to observe the effect of changes in the draft or stoker speed. The pointer is constantly moving and responds to every change in header pressure and rate of feeding the boiler.

Most interesting results are observed in the effect of opening the feed valve. A boiler steaming at 1,000 hp. can be checked down to 300 or 400 hp. for a short time by opening the feed valve. It is not generally appreciated how sensitive a boiler is to this influence, and it astonishes the fireman to see what happens when he suddenly opens up the water supply.

The upper right-hand instrument is a recording thermometer placed in the boiler-breeching connection. This thermometer indicates continuously the temperature of the flue gases and

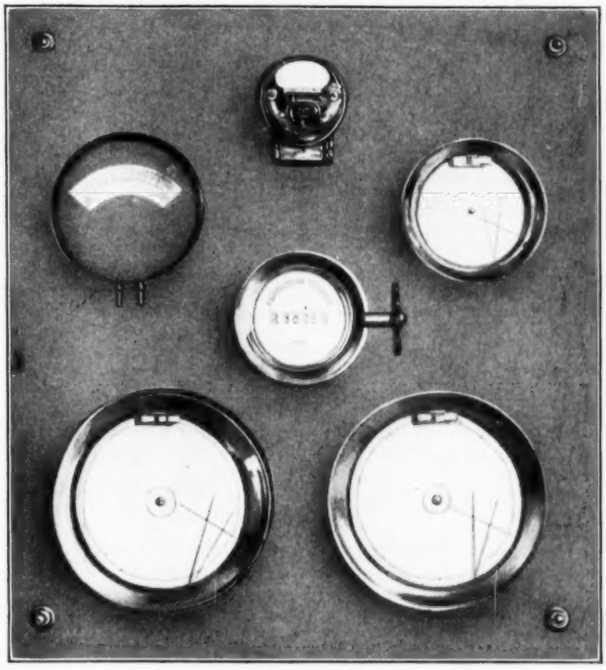


FIG. 1. INSTRUMENTS CONNECTED WITH ONE BOILER

also very clearly the effect of excess air and gas volume. An increase of 50 per cent. in the flue temperature can be made by decreasing the CO_2 2 or 3 per cent. This effect is not generally considered when the desirability of increasing CO_2 is discussed, yet under the conditions obtaining at the plant described, each 100 deg. rise in flue temperature accounted for approximately 3 per cent. of the coal fired. Moreover, this thermometer indicates the condition of cleanliness of of the flues. It is of great value in indicating the boiler efficiency and, when the proper conditions are established and the lowest practical flue temperature once determined, serves as a constant indication whether things are as they should be.

At the middle of the panel is the revolution counter, which is attached to the main driveshaft of the stoker and indicates the number of times the plungers operate. In a stoker of this particular type it is useful in showing uniformity of stoker operation and, as ratios are established, gives the engineer a check on the coal consumption. Inasmuch as the displacement per stroke is nearly constant, a close idea of coal consumption for short periods can be obtained by the indications of the revolution counter.

The large gage at the lower right-hand corner is a Foxboro recording draft gage and is equipped with two pens, the one near the outside of the chart indicating the pressure above atmosphere in the wind box of the stoker and the one nearer the center of the chart indicating the draft, or vacuum, in the furnace.

The boiler was out of commission when the photograph was taken and the two pens stood close together at the zero line

but are shown separated to make the illustration clearer. The range is 6 in. of water pressure and 0.5 in. of water vacuum. These indications show not only what the actual conditions are, but the distance between the two pens indicates very exactly the condition of the fuel bed. If the fuel bed is thin or has a hole in it, its resistance will decrease, and the draft in the furnace will go down, and the distance between the two pens will decrease. With other conditions constant this can be very closely checked against the amount of air per pound of coal or the CO_2 by analysis.

To afford a more continuous and simple check the lower left-hand gage is employed. This is a similar instrument, except that it shows vacuum with both pens; the one nearer the outside shows the vacuum in the furnace, and the other shows the vacuum just inside the boiler damper. This is a direct indication of the loss in the boiler and of the volume of gases passing through the setting.

Furnace-gas volume may be increased in two ways—first, by increasing the rate of combustion and consequently the number of pounds of air required per unit of time, and second, by the infiltration of excess air throughout the fuel bed, furnace door or other openings. The amount of draft loss in the boiler does not indicate anything about the quality of the gases, but is only an indication of the quantity. Other conditions must be considered to determine whether an increased boiler resistance is an indication of efficiency or capacity.

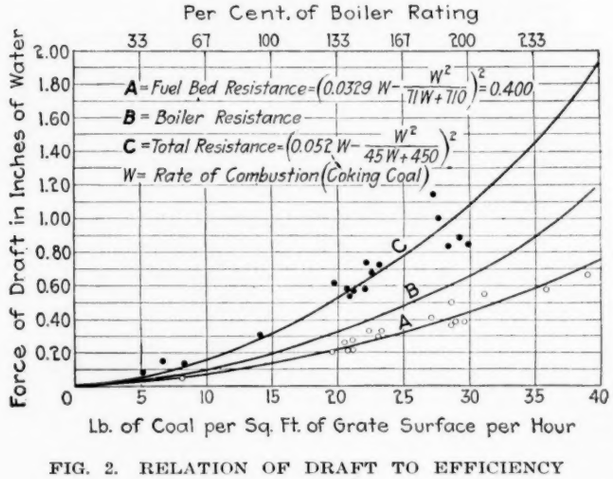


FIG. 2. RELATION OF DRAFT TO EFFICIENCY

If the gas volume is great, as indicated in the boiler-resistance gage, it should mean a high rate of combustion. To obtain a high rate of combustion with the minimum weight of air per pound of coal, a high fuel-bed resistance is necessary, and the pens of this instrument also must stand far apart. With this indication of high capacity is the indication of the flow meter, which proves conclusively that the boiler is actually steaming as indicated by the resistance described.

When a condition like this is established, a definite gas temperature should be obtained, and it soon becomes easy to determine whether the boiler is getting dirty or the baffles getting leaky, by the recorded flue temperature.

Suppose the conditions change and the operator allows the grate surface to become bare; what will be the indications thereof? On account of the wind pressure not having been intentionally reduced, the decreased fuel-bed resistance will have a tendency to bring the furnace draft down toward zero. The two pens of this instrument will approach, and there will be a large volume of excess air passing through the fuel bed and into the furnace. This will result in an unusually large volume of gases for the boiler to take care of, and the resistance will therefore increase, and the two pens of the other draft gage will separate. Where the recording flow meter is in use, the result will be instantly reflected in decreased steam output and if the excess air is not too great, the flue temperature will go up and there will be a loss due both to excess air and improper absorption.

Or assume the reverse of the preceding, where the fuel bed is allowed to become unduly heavy and great resistance is offered to the flow of air through it. The vacuum in the furnace therefore increases and the fuel-bed pens separate, the resultant small volume of gas goes through the boiler with but slight difficulty, and the boiler pens approach. This indicates low capacity and too much CO_2.

Another valuable instrument is a water meter for measuring the boiler-feed water, that not only indicates, but records the conditions of flow and responds to every change in the feed valves. It is interesting to note from its chart that a 2½-in. feed line supplying a 500-hp. boiler, about 25,000 lb. of water per hour, operating at normal conditions can be made to supply by a single turn of the valve handle, 150,000 lb. per hr. It is only by the exercise of great care that changes in the rate of feeding of 20,000 lb. per hr. are obtained, and the results certainly appear to form an argument for the use of automatic feed-water regulators.

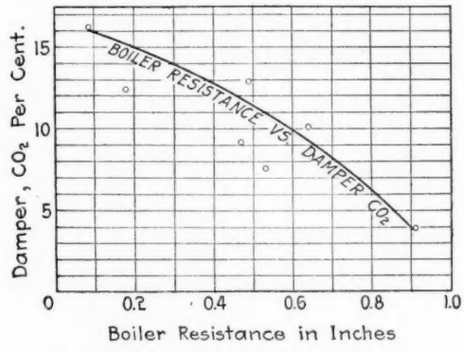


FIG. 3. CO_2 INFLUENCED BY BOILER RESISTANCE

That there is a definite relation between the draft indications and the efficiency of the unit is established. The time available since the first operation of the plant just described did not permit of working out the curves as completely as desired, and we therefore illustrate the principle by curves, Fig. 2, developed at another plant, wherein the boiler was very similar, although the stoker was different. These are curves of constant CO_2, this being around 12 per cent. at the damper of the boiler. The drafts in this case were plotted against the pounds of coal per square foot of grate surface, and the rating of the boiler.

It will be noticed that the points are quite well defined and the curves of such a nature that it is possible to develop an equation from them. Curve A is the fuel-bed resistance—that is, the difference in pressure above and below the fuel bed. Curve B is the boiler resistance, being the difference between the furnace and the damper drafts, and C is the total resistance, being the sum of the other two. It is evident from this that if it was desired to get double rating from this unit, 30 lb. of coal per square foot of grate surface per hour would be required, with a total draft loss of 1.1 in. If it were a natural-draft installation throughout, this would mean that the draft inside the damper would have to be 1.1 in. and the draft over the fire 0.45 in.

In a forced-blast installation, the double pen record would be called into play and the difference between the pen indications utilized to determine the same conditions of resistance.

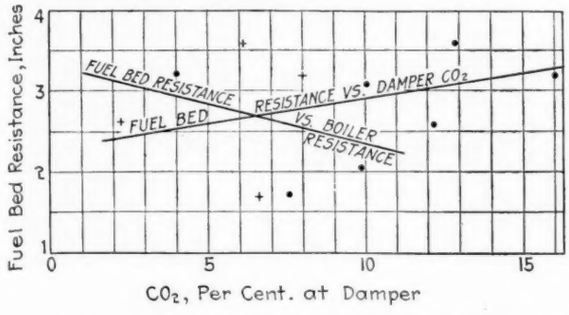


FIG. 4. FUEL AND BOILER RESISTANCE AFFECT CO_2

Under these conditions it would be known that the gases at the damper would analyze 12 per cent. of CO_2.

Should it be desired to operate this boiler at its normal rating, all that would be required would be to drop the damper draft to 0.3 in. and the furnace draft to 0.14 in. in a natural draft, and the result would be obtained.

In how many cases, under ordinary operating conditions without instruments, would it be possible to alter the rate of steaming 100 per cent. with any assurance that efficient conditions would be obtained, unless rather elaborate analyses were taken at frequent intervals? By this method the simple indications of the draft gage are all that is required.

Figs. 3 and 4 indicate the probable effect of changes in draft resistance on the CO_2.

Practical Tests of Steam-Flow and Water Meters

At the recent convention of the National District Heating Association a paper was presented by E. G. Bailey, describing the methods by which Harvard University buys steam on the heat-unit basis. The outstanding feature following the general description of the installation is the close agreement between the amount of steam flowing into the heating system as shown by the flow meters and the pounds of water returned as shown by the V-notch weir meter on the returns.

The accompanying tables show the amounts involved and the percentages of the meters' capacity of steam flowing at the time of the various tests.

A summary of a 24-hr. makeup test on the Harvard heating system, Dec. 3, 1915, at the Harvard Power Station, Boston Elevated Street Ry. shows that all returns from the Harvard yard passed through weir meter 105, and direct to the sump tank. Makeup water passed through a Crown water meter and weighing tank to the sump tank, thence to No. 10 boiler and all steam from this boiler passed through steam meters 90 and 91 in series to the Harvard yard system at 162 to 163 lb. pressure and also through a third steam meter, 108, in the 10-in. line at 50 lb. pressure. The total steam by meters Nos. 90, 91 and 105 respectively, plus allowance for steam for 6 hr. on the mains only when flow meters were not used, was 306,800, 310,800 and 305,800 lb.; total water as metered and delivered to boilers, 310,087 lb.

TABLE 1. RESULTS OF CALIBRATION TESTS OF BAILEY FLUID METERS ON HARVARD HEATING SYSTEM

Meter No.	90	91	90 and 91	90	90 and 91	90	90	90	90	90
Date of test	Nov. 19, 1914	Nov. 19, 1914	Nov. 19, 1914	Nov. 20, 1914	Nov. 20, 1914	Nov. 25, 1914	Nov. 25, 1914	Nov. 23, 1914	Mar. 5, 1915	July 28, 1915
Length of tests, hr	6	6	6	4	6	5	6	7¼	6¼	15
Average steam pressure, lb.	168	168	167.6	166.7	116.0	174.5	173.6	173	175	176
Average steam temp. F.	403	403	396	383.5	380.5	389.3	393	386	400	381
Total steam by meter, lb.	123,660	123,120	127,270	31,620	150,100	29,130	65,200	87,655	78,000	18,885
Total water weight to boiler, lb.	124,030	124,030	129,715	31,145	150,390	30,670	63,895	87,185	77,440	4,850
Difference between meter and weight	—370	—910	+1,555	+475	—290	—1,540	+1,305	+470	+560	.....
Percentage difference	—0.3	—0.7	+1.2	+1.5	—0.2	—5.0	+2.0	+0.5	+0.7	.....
Per cent. of meter, capacity	81	81	42	31	61	22	42	47	46	5

TABLE 2. RESULTS OF CALIBRATION TESTS OF BAILEY WEIR METERS ON HARVARD HEATING SYSTEM

Meter No.	106	106	105	106
Date of test	Dec. 31, 1914	Jan. 28, 1915	July 28, 1915	
Length of test, hr.	5	5.43	15	
Total water by meter	91,200	52,100	21,900	19,100
Total water by weight	91,175	52,190	18,575	
Difference between meter and weight	+25	—90	+3,325	+525
Percentage difference	+0.03	—0.17	+15.2	+2.7
Per cent. of meter capacity	36	19	2.5	

TABLE 3. OPERATING DATA FROM HARVARD UNIVERSITY HEATING SYSTEM

As Reported Each Month Since the System was Placed in Service

Month	Weights			
	Steam Supplied, Lb.	Water Returned, Lb.	Difference, Lb.	Difference, Per Cent.
December, 1914	13,753,070	5,343,000	*	*
January, 1915	13,858,680	11,593,000	*	*
February	12,518,430	12,011,500	506,930	4.0
March	12,758,910	11,259,000	1,499,910	11.8
April	7,487,580	6,368,000	1,119,580	14.9
May	4,369,880	4,090,500	279,380	6.4
June	2,416,150	2,375,000	41,150	1.7
July	867,340	853,670	13,670	1.6
August (13 days)	602,260	592,830	9,430	1.6
September (5 days)	1,082,600	940,550	142,050	13.1
October	7,101,900	6,292,000	809,900	11.4
November	11,050,600	9,985,000	1,065,600	9.6
December	14,145,920	13,375,000	770,920	5.5
January, 1916	14,741,200	14,614,000	127,200	0.9
February	14,865,550	14,943,000	+77,450	+0.5
March	16,111,210	14,991,000	1,120,210	7.0
April	10,640,240	9,484,000	1,136,690	10.6

* Return piping not completed.

Water Power in Sweden—It is estimated that the waterfalls in Sweden represent 5,000,000 turbine horsepower, whereof less than 1,000,000 is at present being utilized. The iron industry makes use of 250,000 hp.; the lumber industry, 260,000; textile factories, 40,000; and plants for the distribution of electric energy, with about 3,000 miles of power-transmission lines, 415,000. Power plants using fuel represent approximately 400,000 hp. About 50 per cent. of all manufactured products are made at factories using water power. —U. S. Commerce Report.

Conference on the Electrical Safety Code

On May 29 and 30, at the Hotel La Salle, Chicago, a conference was held to discuss and perfect the Electrical Safety Code now being prepared by the National Bureau of Standards. Doctor Rosa represented the bureau. The National Electric Light Association and the American Institute of Electrical Engineers were officially represented, as well as a number of large operating companies throughout the country. The two leading associations in the field did not approve, nor would they indorse or ratify the code in its present form. There was a feeling that it went too much into detail and that it would be impossible to apply so minute specifications in all parts of a country where conditions varied so widely. Few states had commissions large enough to see that its requirements were followed. The mediocre ability of the average inspector was another drawback, and standardization in a rapidly developing art would tend to discourage initiative. To determine by definition something which was indefinite by reason of the varying conditions was in their opinion an impossibility. Besides, the code would call for large capital expenditures without adding materially to safety. The two associations would be glad, however, to coöperate with the bureau to formulate a simple clear code that would meet the ends desired.

Similar expressions came from representatives of the operating companies. It was apparent that they did not want their construction specified. Some of them were exceeding the requirements in the code, but they wanted a free rein to build as they saw fit. The question of dollars rather than public safety was uppermost. These general attacks were resented by Doctor Rosa. He pointed out that the code had been perfected at some thirty different conferences in various parts of the country. Suggestions from many companies had been received and sifted, and many of them had been incorporated in the code. Its preparation was a growth rather than a direct product of the bureau. It has been developed through the coöperation of the best engineers in the country. Conditions in different parts of the country had been taken into account as well as various kinds of service and the interests of the companies concerned. A careful reading would show that the requirements are modest and, in fact, under some of the construction of the bigger companies. A code too general would be vague and indefinite. It is time for specific criticisms rather than general attacks. It is not a question of saving a little money for operating companies, but the safety of all concerned.

Doctor Rosa made a plea that the code be given a year's trial, the various companies complying with it as nearly as possible without imposing hardships. In this way inherent defects would show up, and at the end of the year conferences could be held to revise or make any alterations that might be deemed advisable. The aim of the code is to come as near 100 per cent. safety as is reasonable and practical. It is the best and most thorough body of rules ever brought together for the purpose and is at least worthy of trial. Its adoption by state commissions rather than by municipal or state authorities was recommended so that changes could be effected more readily.

Following these general remarks the code was read paragraph by paragraph, the discussion dwelling mostly on line sag, concentric wiring, towers and strength of construction at crossings of high-tension lines with railways and telephone lines. With the reading it became evident that the code was not at all drastic and not nearly so impracticable as had been implied. When the general prelude is included, it is broad enough to meet all conditions of climate and service without imposing hardships on any of the operating companies. However, the operating men stuck to their guns and no definite agreement was reached. It is likely that some of

the suggestions received at the conference will be incorporated in the code and it will be published officially soon after. It may then be adopted at the discretion of the various state commissions.

Annual Convention American Association of Engineers

On May 9 and 10, in the rooms of the Western Society of Engineers, Chicago, Ill., the American Association of Engineers held its first annual convention. An interesting point in the secretary's report was the announcement of a membership of 700. Since the first of the year an average gain of about 100 new members per month has been made. One of the features of the convention was the passing of a resolution offering to the Government the services of the association in helping to advance military preparedness. The resolution asked for information as to the best methods of coöperation and included a request for blanks on which the various members might enter their specific acquirements and training. At the banquet, held Tuesday evening, some phase of preparedness was touched upon by such speakers as Colonel Allen, Alderman Utpatel and Dr. Vernon Hill, of Chicago; Captain Reilly, of the "Tribune" and G. W. Rich, an attorney of Rochester, N. Y.

For the ensuing year Garrison Babcock automatically becomes president of the association. The following officers were elected: W. J. Mauer, director; B. Moreel, Jr., T. J. Mullin and C. A. Guensseln, advisory committee. A number of standing committees were also appointed.

New Jersey N. A. S. E. Convention

The State Association of the National Association of Stationary Engineers in New Jersey held its twenty-fifth annual convention in the Auditorium, Paterson, May 29 to June 4. The exhibit at the Auditorium was open all the week, while the business session was held Saturday and Sunday in the rooms of the Republican Club. The Auditorium, formerly an old silk mill, is nearly 500 ft. long by approximately 40 ft. wide with no columns, and made excellent space for the sixty exhibitors.

Monday evening, May 29, Mayor Amos H. Radcliffe, who is a member of the local association in Paterson, opened the exhibit and welcomed the convention. Attendance at the exhibit was fairly good, and on Saturday, when delegates began arriving from other cities throughout the state, it was excellent.

Saturday afternoon at 2:30, State President Dennis F. Bartley, of Jersey City, called the delegates to order and received preliminary reports. National President Walter H. Damon, of Springfield, Mass., addressed the delegates, speaking particularly on the business methods of the association, stating also that he believed the association would lose the suit in which it was involved with a company that formerly printed the association's paper, the "National Engineer." Without other important business the convention adjourned to meet the next morning at 10 o'clock.

Saturday evening the delegates and their friends enjoyed a smoker at the Republican Club rooms, entertainment being given by "the bunch," which was joined by its old star singer, Frank Corbett.

Sunday the various committees reported. Charles H. Bromley, of the state educational committee, reported little work as compared with that of the few years previous, when vigorous educational campaigns had been conducted by his committee. The association voted $100 for educational work for the coming year. Mr. Bromley detailed the educational methods used in the Newark association, of which he is instructor, and urged their adoption by other associations.

Frank A. La Point, of Jersey City, chairman of the legislation committee, made a complete report on the efforts to secure amendments to the engineers' and firemen's license law. One of these amendments was the increasing of the license bureau examiners' salaries from $1,200 per year to $2,000. Governor Fielder vetoed the bill carrying this amendment. The convention voted $150 for legislative work for the coming year. The recommendation of the ways and means committee that the members be assessed thirty cents each was carried.

Considerable discussion was had on the question of biennial national conventions instead of annual, as now practiced, those favoring the former arguing that the cost of the annual conventions deplete the treasury every year, and that biennial meetings would correct the evil. Those defending the annual-convention plan feared loss of interest and coöperation by the association's friends and patrons if national conventions were two years apart. Emphasis was laid on the importance of selecting the convention city in or near the center of membership, which is Cleveland, Ohio. The next national convention will be in Minneapolis, Sept. 11 to 16, and National President Damon stated that instead of the payment of the usual five cents per mile to delegates, three cents would be the limit. The convention went on record as favoring the present annual national convention plan.

John J. Reddy, state deputy, reported the institution of two new associations (Bayonne and Phillipsburg).

The officers elected are: Thomas Brown, Newark, president; John J. Reddy, Jersey City, vice-president; James S. Heath, Elizabeth, secretary, reëlected; William Krauss, Passaic, treasurer, reëlected; Allen Kenney, Jersey City, conductor, reëlected; V. Secor, Phillipsburg, doorkeeper; state deputy, M. P. Kelley, Paterson.

The next convention will go to Hoboken, and it is likely that that city will "break the ice" in eliminating the large exhibits such as have characterized the last several New Jersey conventions. This change will be appreciated by the small cities, the delegates from which would like to welcome the convention, but do not care to assume responsibility for a large exhibit. There is little doubt that most of the exhibitors will welcome the change.

ENGINEERING AFFAIRS

The American Society for Testing Materials will hold its nineteenth annual meeting at Atlantic City, N. J., June 27-30, with headquarters at the Hotel Traymore.

The Southwestern Electrical and Gas Association at its recent convention in Galveston, Tex., elected the following officers: President, E. R. Slater, of Dallas; first vice-president, H. C. Morris, of Dallas; second vice-president, D. A. Hegarty, of Houston; third vice-president, W. A. Sullivan, of Shreveport; secretary, H. S. Cooper, of Dallas; treasurer, J. B. Walker, of Dallas.

OBITUARY

P. A. Moulton, New York agent of the Sims Co., Erie, Penn., died suddenly May 24 while on the train bound for Albany.

MISCELLANEOUS NEWS

A Municipal Electric-Light Plant is to be installed this year at Imogene, Iowa.

Transmission Line to Creston, Iowa—The citizens of Orient, Iowa, have voted $10,000 bonds for the installation of a municipal electric-light transportation line from Orient to Creston, Iowa.

Electric Plant at Norden, S. D.—Construction of an electric light and power plant and water-works system is to be undertaken this season at Norden, S. D., according to R. C. Byrde, town clerk.

New Plant at Perry, Iowa—The Iowa Railway and Light Co. plans to erect a $100,000 power plant at Perry, Iowa, from which to furnish electric current to Grand Junction, Dana, Paton, Bouton, Woodward and Coon Rapids.

Alabama Power Co.'s New Plant—Construction work on the new $2,000,000 steam auxiliary power plant of the Alabama Power Co., to be located near Drifton Station on the Warrior River in Walker County, will begin within a few weeks, according to information obtained at the office of the power company.

Power-Plant Additions in Wisconsin—The De Pere Electric Light and Power Co. plans to remodel its power plant at De Pere and to change the current output from direct to alternating. The contemplated improvements are estimated to cost $20,000. L. L. Tessler is superintendent. The Milwaukee Light, Heating and Traction Co., headquarters at Milwaukee, is preparing plans for a new boiler house and two new 50-hp. boilers for a plant at Racine. John Anderson is chief engineer.